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Largest Oil Tankers in the United States

Description of First of Two Large Oil Tank Steamers Built for the Standard Oil Company at Newport News, Va.

The tank steamer *John D. Archbold*, whose speed trials took place on March 11, and a duplicate oil tanker to be named the *John D. Rockefeller*, now on the ways of the Newport News Shipbuilding and Dry Dock Company, Newport News, Va., have the distinction of being the largest tankers built or building in the United States. The contract for these vessels was placed with the Newport News yard by the Standard Oil Company of New Jersey.

The *John D. Archbold* was built under the special survey of Lloyds Register of British and Foreign Shipping to Class A-1 for shelter deck, longitudinally-framed vessels. She is designed to carry petroleum in bulk, and the shelter deck is fitted to provide sufficient strength and freeboard. Panama and Suez Canal certificates have been issued to the vessel. The accompanying plans and midship section illustrate the general arrangement and structural scantlings. The principal dimensions are as follows:

Length over all	474 feet 6 inches
Length between perpendiculars (Lloyds)	460 feet
Beam, molded	60 feet
Depth, molded to shelter deck	36 feet 2 inches
Depth, molded to upper deck	28 feet 8 inches

The designed deadweight capacity is 10,000 tons, which includes cargo, fuel, fresh water, stores, etc., on a mean draft not exceeding 23 feet 4 inches in salt water, with a coefficient not to exceed .805. The vessel is arranged to trim on an even keel with a draft of 22 feet when carrying a total deadweight of 9,000 tons, 8,600 tons of which will be cargo oil in the main tanks, the remaining 400 tons consisting of fuel, water and stores.

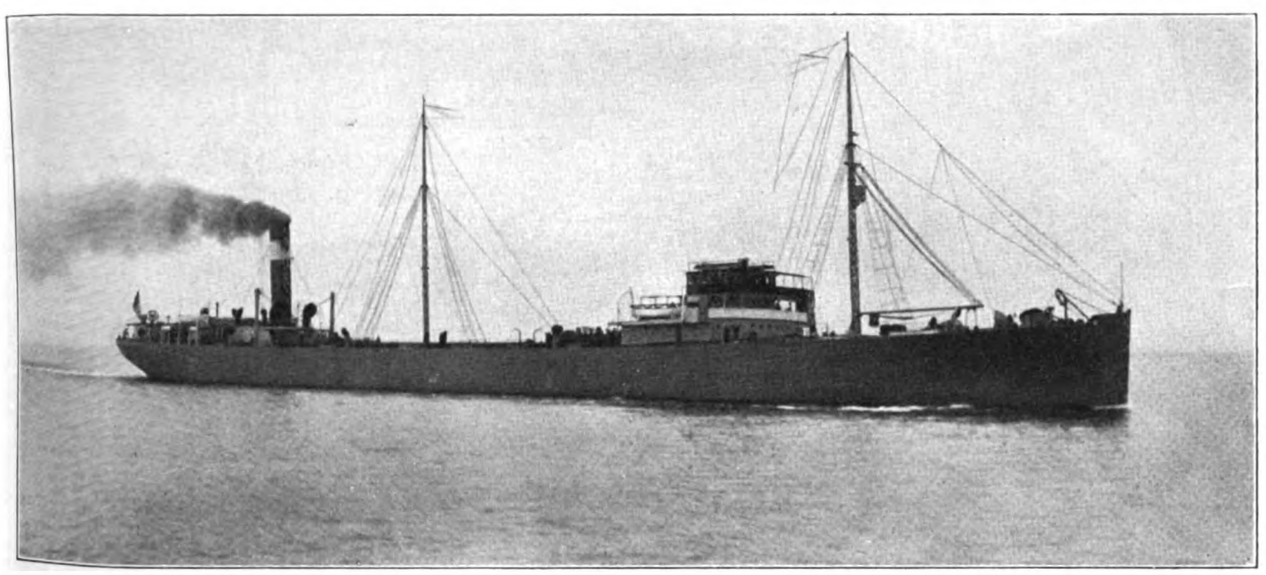

Fig. 1.—Oil Tank Steamer *John D. Archbold*, of 10,000 Tons Deadweight Carrying Capacity

General Arrangement

The vessel is built with longitudinal framing. The hold is divided into nine tanks for carrying cargo oil with a center-line bulkhead extending to the upper deck, dividing these tanks into port and starboard compartments. The spaces in way of the expansion trunk at the sides of the ship are divided into four compartments on each side to be used for summer oil tanks. The hold forward of the cargo tanks, and the space forward of the coal bunker under the shelter deck are for carrying case oil, or other package freight. Cofferdams are fitted between the cargo holds and the oil tanks at the forward end, and between the fuel oil tank and the cargo oil tanks aft. The pump room is located amidships, and there is a deep tank under the forward hold. The machinery is located aft, and a double bottom is fitted under the fuel tank and the machinery spaces.

There are three decks, the main, upper and shelter. The space under the shelter deck, abreast the boiler casing, extending forward to amidships, is arranged for use as a coal

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bunker. The fuel tank is arranged for carrying either coal or oil fuel, although for the present the vessel will use coal for fuel. The main and summer tanks and the fuel oil tanks extend to the upper deck only, and the space at the hatches between the main and shelter decks is bulkheaded off from the rest of the shelter deck space to prevent fumes from settling under the shelter deck.

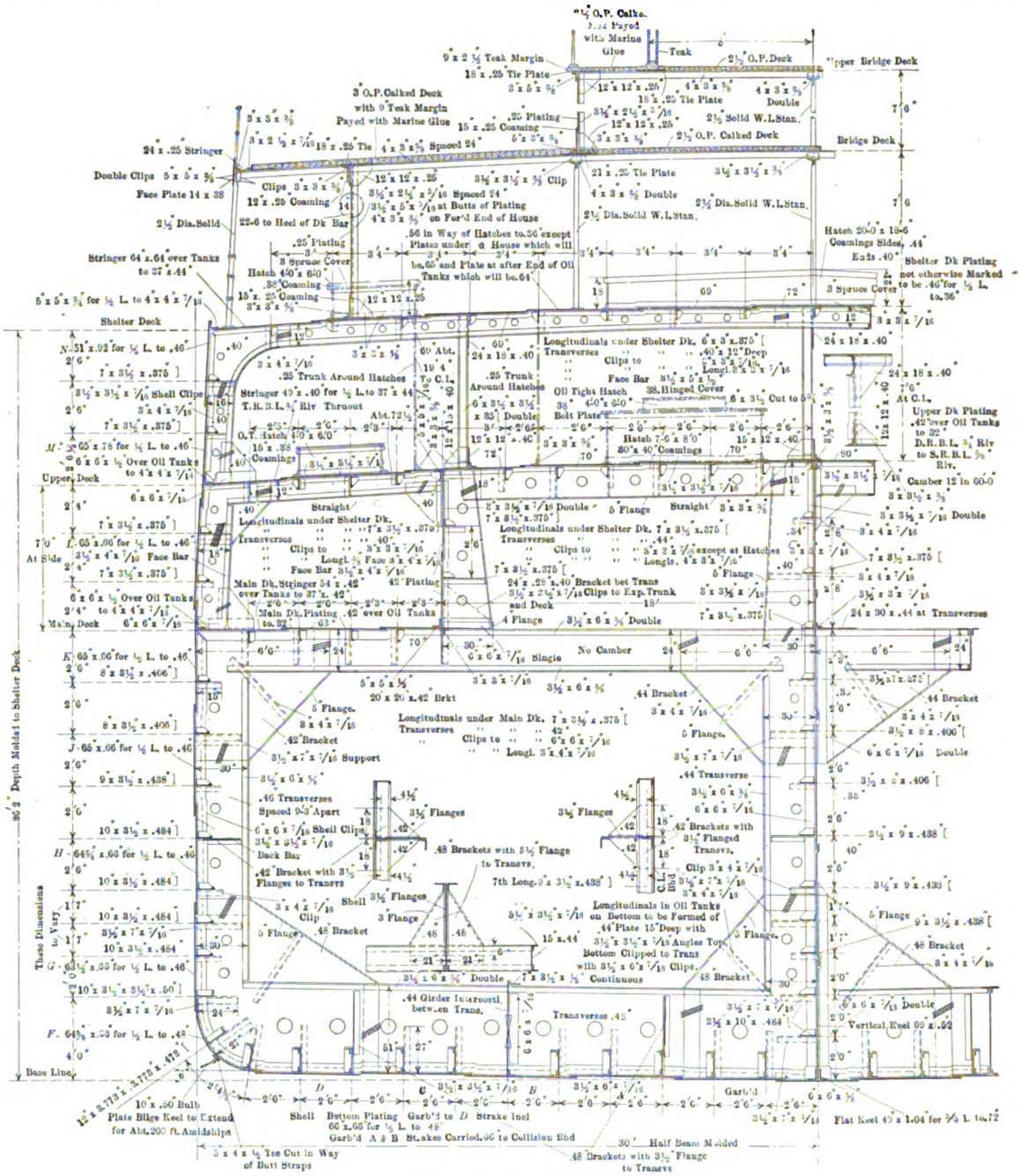
Upper Bridge Deck
Bridge Deck
Shelter Deck
Upper Deck
Main Deck
Base Line
Depth Molded to Shelter Deck
These Dimensions to Vary
Flat Keel 45 x 1.04 for ½ L. to .72
Vertical Keel 60 x .62
Garb'd
Shell
Bottom Plating
Garb'd A & B
Half Beam Molded
Transverses .48
Longitudinals in Oil Tanks on Bottom to be Formed of .44 Plate 15" Deep with 3½" x 3½" x 7/16 Angles Top & Bottom Clipped to Trans with 3½" x 6" x 7/16 Clips. 48 Bracket
Longitudinals under Shelter Dk. 6" x 3" x .375" [
Longitudinals under Shelter Dk. 7" x 3½" x .375" [
Longitudinals under Main Dk. 7" x 3½" x .375" [
Hatch 20-0" x 15-6"
Coamings Sides, .44
Ends .40"
Shelter Dk Plating not otherwise Marked to be .40 for ½ L. to .56
3 Spruce Cover
Upper Dk Plating .42 over Oil Tanks to 32"
Camber 12" in 60-0
No Camber
Straight
Double
Single
Bracket
Flanges
Transverse
Support
Brkt
Plate
Bar
Clip
Bulb
Tee Cut in Way of Butt Straps
Plate Bilge Keel to Extend for Abt. 200 ft. Amidships
Strake Incl
Stakes Carried .66 to Collision Bhd
Brackets with 3½" Flange to Transvs

Fig. 2.—Midship Section

The rigging consists of two pole masts, schooner-rigged. Accommodations for the officers, purser and doctor, together with the pantry and dining saloon, are located in a deck house amidships, while the galley is on the shelter deck aft. The engine room crew, firemen, sailors, quartermasters and bos'n have quarters under the shelter deck aft.

The equipment of the vessel is very complete, including a submarine signal receiving set, and wireless apparatus with accommodations for two operators. Lifeboats with a total capacity which is double the number of persons on board are provided, consisting of two 24-foot steel boats and two 24-foot wooden boats

MACHINERY

The propelling machinery consists of one 4-cylinder quadruple-expansion engine, with cylinders 24, 35, 51 and 75 inches diameter by 51 inches stroke. The high-pressure and first intermediate-pressure cylinders are fitted with piston valves, while the second intermediate and the low-pressure cylinders are fitted with double-ported slide valves. All of the valves are controlled by Stephenson link motion. The reverse gear is of the direct-acting type, driven by a reversing cylinder 14 inches diameter by 24 inches stroke.

The bedplate of the main engine is of the cast iron box type, made in sections, with eight main bearings fitted with water circulators and lined with white metal; the top bearing is of cast steel in one piece. The crank shaft is of the built-up type with interchangeable sections. The propeller is of the four-bladed sectional type with manganese bronze blades and cast iron hub. The diameter is 17 feet 6 inches and the pitch 16 feet 6 inches.

An independent condenser of the cylindrical type with steel

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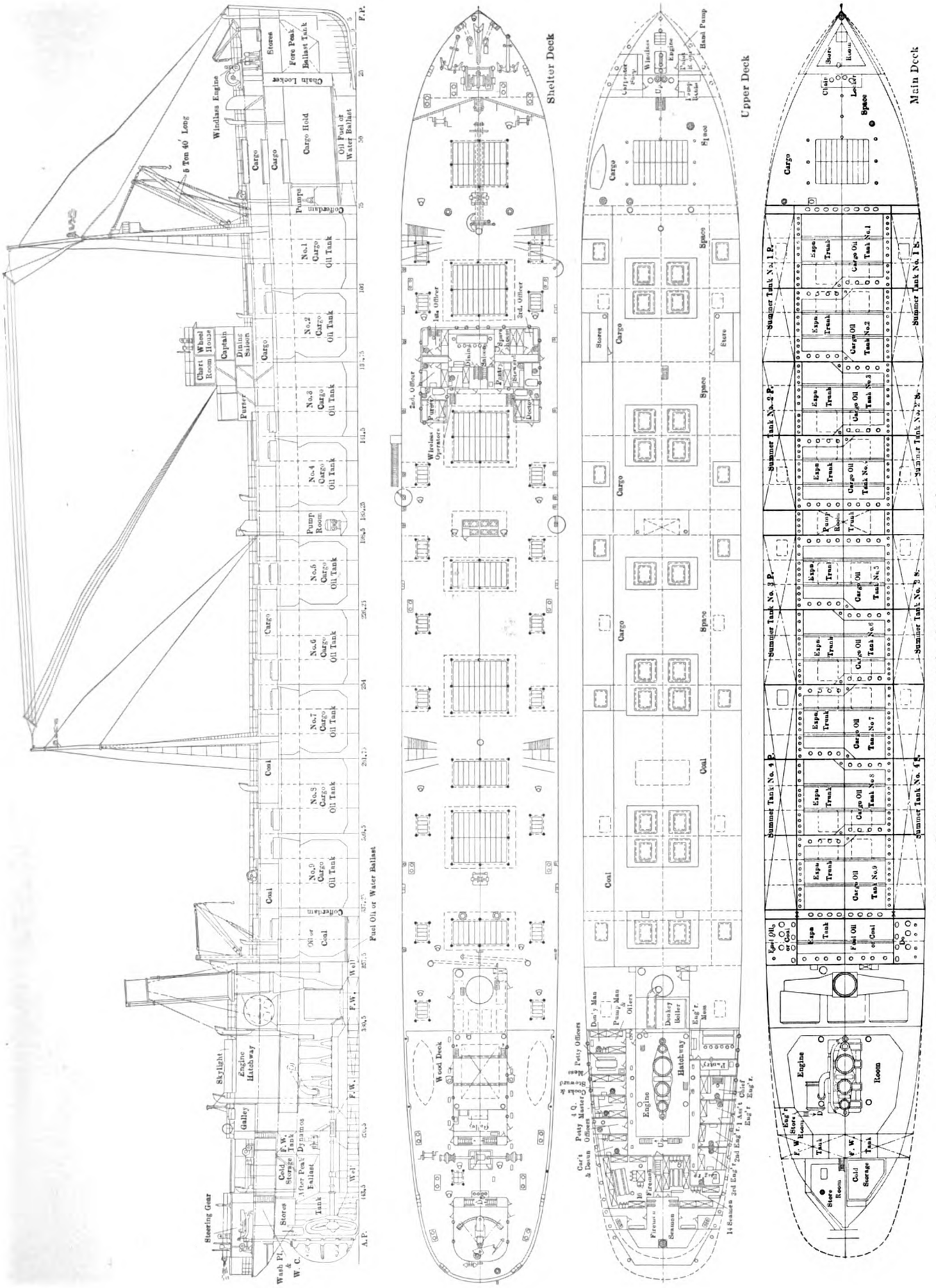

Fig. 3.—Profile and Deck Plans of the Oil Tank Steamer *John D. Archbold*

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shell and Muntz metal tube plates and ¾-inch Admiralty mixture tubes provides a cooling surface of 4,200 square feet. The air pump is of the Edwards type, 26 inches diameter and 24 inches stroke. Cooling water is supplied by an independent centrifugal pump with two direct-acting engines, one of which will do the work, the other standing by as a spare.

Steam is supplied at a pressure of 220 pounds per square inch by three single-ended Scotch boilers, each 14 feet 4 inches mean diameter and 11 feet 6 inches long, arranged for Howden's system of forced draft. The combined heating surface of the boilers is 7,035 square feet, and the total grate surface 177 square feet. Each main boiler is fitted with Stewart circulators. As previously stated, the boilers will be coal-fired at present.

A donkey boiler, 10 feet 11 inches inside diameter by 10 feet 10½ inches long, is located at the main deck in the boiler casing. The heating surface is 1,223 square feet, and the grate area 39 square feet. The boiler is designed for a working pressure of 180 pounds per square inch.

Two duplex National Transit oil pumps, 18 inches by 14 inches by 24 inches, are connected to the cargo pumping system. The cargo tanks are fitted with fire extinguishing connections and heater coils.

Two General Electric vertical engine generating sets of 20 kilowatts capacity each furnish light for the vessel, and current for the ventilating fans. The deck machinery includes a Hyde windlass with quick warping drums, three warping winches with extra large gypsy heads and a Brown tiller.

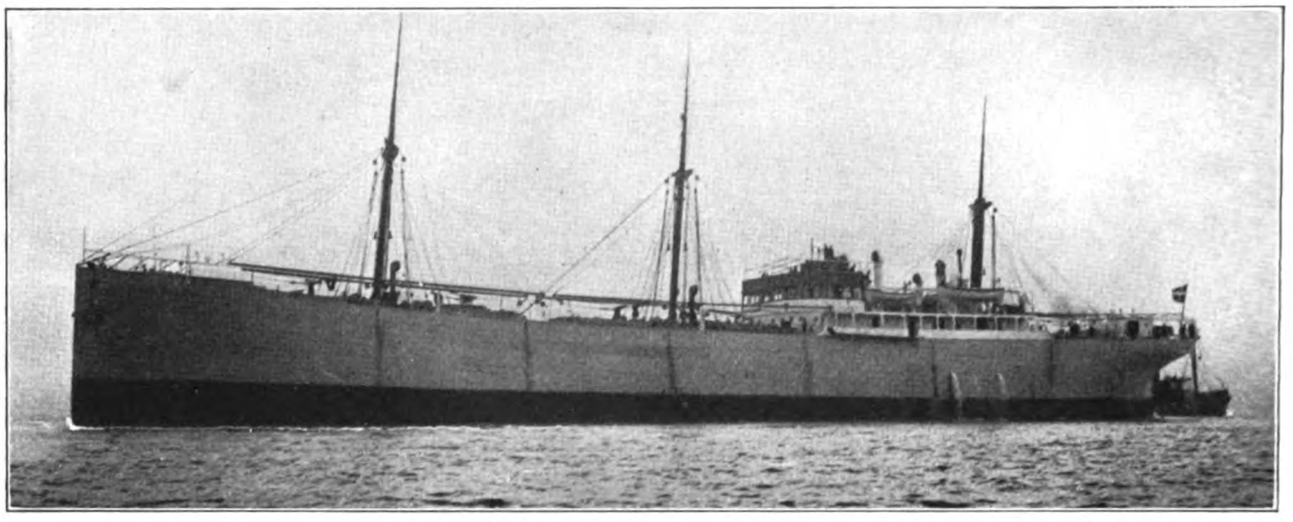
On trial, a speed of 12.3 knots was maintained, the engines developing 3,400 indicated horsepower. It is stated that the power was developed with a very low coal consumption.

Lake Launchings

The steamer *D. W. Crawford,* building for the Virginia Steamship Company, was launched April 18 at the Lorain yard of the American Shipbuilding Company, and the steamer *W. H. Donner,* building for the Mahoning Steamship Company, was launched May 2 at the Ashtabula yard of the Great Lakes Engineering Works. Both of these ships are duplicates in all essential details and of the following dimensions: Length overall, 524 feet; keel, 504 feet; beam, 54 feet; depth, 30 feet. They are built upon the Isherwood system of longitudinal framing, and will carry in the neighborhood of 9,000 tons. Triple-expansion engines are installed having cylinders 23½ inches, 38 inches and 63 inches diameter and a stroke of 42 inches. Two Scotch boilers, each 15 feet, 4½ inches mean diameter by 12 feet long overall, built for a working pressure of 180 pounds per square inch, will supply steam for all purposes. Both of these ships replace vessels lost in the storm of November last, and they have been designed with all possible care and with the experiences of that storm as a guide; hence they should prove very stable and satisfactory ships for their intended service. They have been built to meet the requirements of the Great Lakes Register and British Lloyd's.

Motor Ship Kronprins Gustaf Adolf

Trials of the motor ship *Kronprins Gustaf Adolf,* the third motor ship built by Burmeister & Wain, Copenhagen, Denmark, for Rederiaktiebolaget Nordstjernan, Stockholm, were carried out in Copenhagen Sound March 15, although, on account of foggy weather, it was impossible to make them complete. After the trials, however, the ship sailed to Frederikstad and Göteborg to take on cargo, and a final trial trip with full load was then carried out on the 24th. The dimensions of the ship are as follows:



Twin-Screw Motor Ship *Kronprins Gustaf Adolf,* Equipped with Two Six-Cylinder Main Engines of 1,000 Horsepower Each

Length	362 feet
Beam	51 feet 3 inches
Depth	23 feet 1 inch
Deadweight	6,550 tons

The new vessel is a sister ship to the *Suecia* and *Pedro Christophersen,* which Burmeister & Wain recently built for the same owners. The engine installation of the *Kronprins Gustaf Adolf* differs from that in the other ships, however, in that the main engines have six cylinders instead of eight, as in the other ships. The two main engines are each of 1,000 brake horsepower, the dimensions being 21.6 inches cylinder diameter, 29.2 inches stroke, and 140 revolutions per minute. In other respects the machinery is the same as in the vessels formerly delivered.

The normal speed of the ship is 10 knots, and the maximum speed 10¾ knots, although on the trial trip a maximum speed of 11.83 knots was attained. The oil consumption worked out at .337 pound of Rumanian crude oil per indicated horsepower per hour for all purposes, including the oil used in the auxiliary machinery, which gives a fuel consumption of .397 pound per shaft horsepower hour.

After the full load trials, the ship proceeded on her first voyage to South America.

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Measurement of Strains in a Ship's Hull*

New Type of Gage Used to Measure Strains in Hull of Ship at Sea and While Receiving Cargo—Results Obtained on Typical Voyage

BY JAMES E. HOWARD†

Strain gage measurements were made on the plates of the shelter deck of the steamship *Ancon*, of the Pacific Railroad Steamship Company, on a voyage from New York to Colon and return, and subsequently during the time the ship was taking cargo aboard at New York.

The *Ancon*, originally called the *Shawmut*, was built by the Maryland Steel Company, Sparrow's Point, Md., in 1902. It has the following dimensions:

Length, 505 feet; beam, 58 feet; depth, 41 feet. Displacement at 30 feet draft, 20,830 tons.

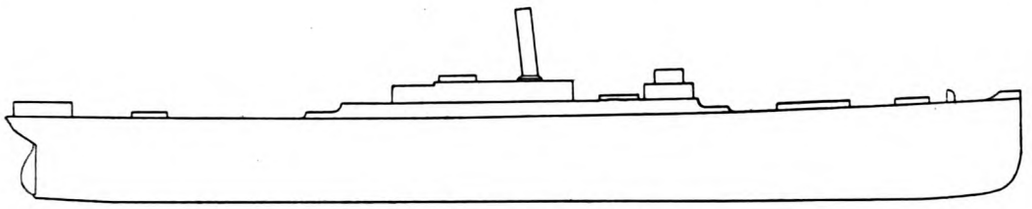

Fig. 1.—Outboard Profile of *S. S. Ancon*

It has twin screws, and is equipped with triple-expansion engines, with cylinders 24 inches, 39½ inches and 62 inches diameter, and 45 inches stroke, making 78 revolutions per minute. The steam pressure carried is 200 pounds, and the ship makes a speed of about 12 knots.

For a time the ship was used in the China trade, but of late she has been used in the transportation of structural materials for the Panama Canal, of which cement usually formed the principal part of the cargo.

Live load strain measurements were made on the deck plating while the ship was at sea, using a new type of extensometer designed by the writer, a scissors gage (Fig. 2), so-called from the resemblance of its working parts to a pair of scissors.

THE STRAIN GAGE

It consists of two beams, which, taken together, are rectangular in outline, and between which are located two small bell crank levers. Parallel motion of the beams is secured by the use of flexible stay plates. The bell crank levers are mounted upon fulcrums made of thin plates of tempered steel. They work in opposite directions. The long arms of the levers are blades which make with each other a small angle, the vertex of which or point of intersection of the blades marks the place where the reading of the instrument is taken.

The blades are comparatively light, the beams are relatively heavy. The instrument is dust proof. The proportions of the parts are such that a change in the position of the point of intersection of the blades of 1 inch in length represents a movement of the beams, or measured movement of one-thousandth of an inch. The index plate is graduated into ten parts, each representing one ten-thousandth of an inch. A change in position of the point of intersection of .01-inch on the scale represents one one-hundred-thousandth of an inch movement of the beams. The gage has a total range of movement, or full scale reading of one-thousandth of an inch.

Since the range of the instrument is quite limited it is necessary to establish longer or shorter-gaged lengths on the measured member, according to the expected magnitude of the strains in order that they shall not exceed the capacity of the instrument. On the present occasion, the strains were generally small, and it was expedient to use extension rods which covered a length of 6 inches on the deck plates. The gage is shown in position on the deck in Fig. 3, where it

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† Engineer-Physicist.

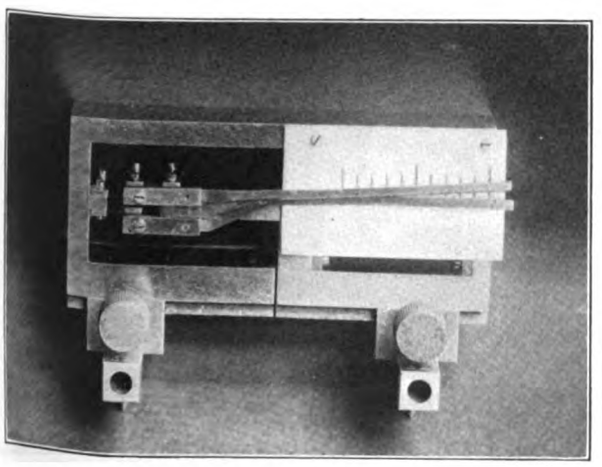

Fig. 2.—Scissors Gage

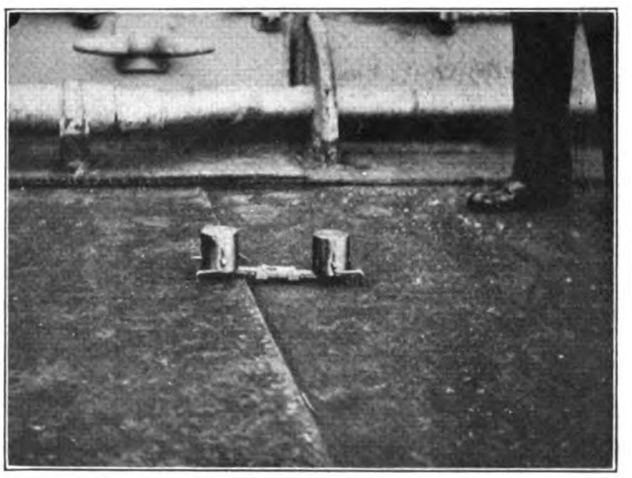

Fig. 3.—Scissors Gage Spanning Lap Joint on Deck Plates

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appears spanning a lap joint, stepping down from one plate to another. Contact with the deck plates is made by means of conical steel points attached to the beams or to the extension rods. Sufficient roughness was found on the deck

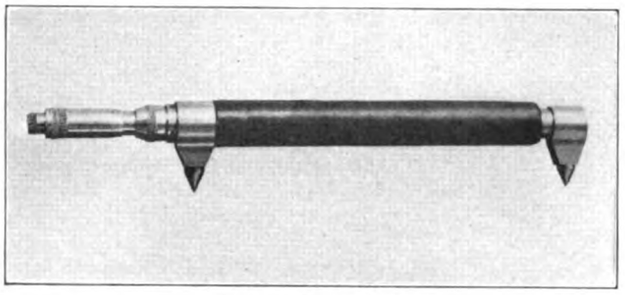

Fig. 4.—Strain Gage

plates to actuate the instrument when a weight was placed over each contact point, as illustrated in Fig. 3. The contact points could be placed in small holes made with a center punch in the deck plates if needed.

Moderate loads causing strains of a few hundred-thousandths of an inch, or slowly applied loads, can be measured with comparative ease, but the effects of rapidly-changing loads of considerable magnitude are less easily observed. The instrument is a new one and was used for the first time on this voyage. A number of modifications in its details will be made to facilitate taking readings, making it even a little more compact, and providing means for attaching it to different-shaped members and in different positions.

It responds promptly to changes in strains in the member under examination. Stamping upon the deck at a distance of 25 feet from the gage caused longitudinal vibrations which the instrument responded to. The results here presented are regarded as close approximations to the actual strains which were developed, but are not held to be rigorously exact.

A strain gage of the type shown in Fig. 4 was used for measuring the deck plates during the time cargo was being received. The gage employed had a working length of 20 inches. It is telescopic in design. There is an outer tube which carries a conical contact point near one end of the instrument, and an inner bar which carries a contact point near the opposite end. The longitudinal movement of the inner bar with reference to the outer tube, and consequently the movement of the contact points, is measured by

TABLE No. 1

STRAINS IN PLATES OF SHELTER DECK OF S. S. ANCON WHEN AT SEA, LOADED WITH 10,217 TONS OF CARGO DRAFT ABOUT 29 FEET

Measurements with scissors gage, lengths of 6 inches each, on deck plating, port side of ship, on outward trip, New York to Colon, April, 1913. Observations on plates at middle of width and length, also stepping down from one plate to another, across triple-riveted lap joints. Position of ship, off easterly end of Cuba. Smooth sea. Forward part of ship.

Course of Plating	Plate		Gaged Length				Probable Effect of	
	No. of	Weight of, pounds	Location	On solid plate or joint	Measured Strain, Inches	Corresponding stress, pounds per square inch	Engines	Pitching
C	27	22	Just aft breakwater	Solid plate	No strain detected			
C	26 to 27	22	Abreast Hatch No. 1, just aft breakwater	Joint	.00006			
C	26	22	" " " 1	Solid plate	No strain detected			
C	25 to 26	22	" " " 1	Joint	.00006			
C	25	22		Solid plate	.00003			
C	24 to 25	22		Joint	.00012			
C	24	22	Abreast Hatch No. 2	Solid plate	.00002	100		
C	23	22	" " " 2	Solid plate	.00007	350	.00005	.00002
C	22 to 23	22	" " " 2	Joint	.00015		.00010	.00005
C	22	22	" foremast	Solid plate	.00013	650	.00008	.00005
C	21 to 22	22	" "	Joint	.00032		.00020	.00012
C	21	22	" Hatch No. 3	Solid plate	.00005	250	.00003	.00002
C	20 to 21	22	" " " 3	Joint	.00015		.00010	.00005
C	20	22	Just aft forward bulkhead of superstructure	Solid plate	.00002	100	.00002	0
C	19 to 20	22	Aft forward bulkhead of superstructure	Joint	.00020		.00020	0
C	19	22	" " " "	Solid plate	.00002	100	.00002	0
C	18 to 19	22	Abreast forward end Hatch No. 4	Joint	.00030		.00030	
C	18	22	Abreast Hatch No. 4	Solid plate	.00010	500	.00010	
C	17 to 18	22	" " " 4	Joint	.00020		.00020	
C	17	22	Abreast, after end Hatch No. 4	Solid plate	.00005	250	.00005	
C	16 to 17	22	Just forward of boiler hatch	Joint	.00030		.00030	
D	23 to 24	22	Abreast forward end Hatch No. 2	Joint	.00006			
D	22 to 23	22	Abreast Hatch No. 2	Joint	.00015		.00010	.00005
D	21 to 22	22	" after end Hatch No. 2	Joint	.00020		.00020	
D	20 to 21	22	" forward end Hatch No. 3	Joint	.00022		.00022	
Stringer	Butt strap to 25	30	" Hatch No. 2	Joint	.00003			
"	Butt strap to 24	32	" " " 2	Joint	.00003			
Stringer	23	22	" " " 2	Solid angle	.00002	100		
				After Part of Ship.				
C	9 to 10	22	Just aft, after bulkhead of superstructure	Joint	About .00125		Abt. .00125	
C	9	22	Aft, after bulkhead of superstructure	Solid plate	.00020	1,000	.00020	
C	8 to 9	22	Abreast forward end Hatch No. 6	Joint	.00090		.00060	.00030
C	8	22	Abreast Hatch No. 6	Solid plate	.00030	1,500	.00030	
C	7 to 8	22	" " " 6	Joint	.00050		.00030	.00020
C	7	22	" " " 6	Solid plate	.00018	900	.00018	
C	6 to 7	22	" after end Hatch No. 6	Joint	.00026		.00016	.00010
C	6	22	Nearly abreast main mast	Solid plate	No strain detected			
C	5 to 6	22	Abreast main mast	Joint	.00010		.00005	.00005
C	5	22	Just forward of Hatch No. 7	Solid plate	No strain detected			
B	8	16	Just aft, after bulkhead of superstructure	Solid plate	.00045	2,250	.00045	
A	7	16	" " " " " " "	" "	.00035	1,750	.00030	.00005
Stringer	Butt strap to 10	34	Aft, after bulkhead of superstructure	Joint	.00015		.00015	
Stringer Angle			Abreast forward end Hatch No. 6	Solid angle	.00025	1,250		
Bulwark rail			Aft, after bulkhead of superstructure	" "	.00026	1,300		
Earlier	Observations made		when a Moderate Sea was Running. Position of Ship, off Watlins Island.					
C	8	22	Abreast Hatch No. 6, 2 feet from coaming	Solid plate	.00050	2,500		
D	8	22	Abreast forward end Hatch No. 6	" "	.00070	3,500		
D	7	22	Abreast Hatch No. 6	" "	.00060	3,000		
Bulwark			Aft, after end of superstructure	Solid angle	.00110	5,500		

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a screw micrometer attached to the outer member. The contact points are inserted in small drilled and reamed holes made in the deck plates, and when thus centered readings with the micrometer are taken.

The gage is used in connection with a steel reference bar, made of the grade of structural steel, and lengths on the deck are referred to this standard bar and recorded in terms of it. Concerning manipulative features, check observations, when the instrument is used under favorable conditions, commonly give the same ten-thousandths of an inch reading of the micrometer. In engineering structures in general it is not expected to define stresses much nearer than 500 pounds per square inch, which corresponds to a strain of three and one-third ten-thousandths of an inch in 20 inches length.

The deck plates of the ship afforded a better opportunity than usually experienced in making strain gage measurements on engineering structures; therefore, it is believed that these results are trustworthy in a higher degree than generally expected, that the strains were substantially as reported. In deducing the stresses corresponding to the measured strains, a value of 30,000,000 pounds per square inch was used for the modulus of elasticity of the steel plates.

A full cargo of 10,217 tons was carried on the outward voyage, while on the return voyage the cargo was only 3,000 tons.

Live Load Strains in the Deck Plating

Observations on the live load strains of the deck plating were made on both the outward and the return voyages. Good weather generally prevailed, hence the measured strains refer to conditions as they were found on a smooth sea with a minimum pitching of the ship. A few observations, however, were made on the outward voyage with a moderate sea running. With a low freeboard, when the ship carries a full cargo, the decks were generally wet in a moderate sea, and without protection it was impracticable to make many observations. The few measurements which were taken showed strains of increased magnitude, corresponding to the greater pitching of the ship which then occurred.

Tables 1 and 2 give the results of the measurements made at sea. Certain of these measured strains are entered upon diagrams of the deck plates and appear in Figs. 5, 6 and 7.

Referring to the observations in detail, and beginning at the forward deck, neither the solid metal of plate *C*, No. 27, over which the breakwater passed, nor that of the next plate aft, developed strains of sufficient magnitude to be measurable. At the first lap joint aft the breakwater, and across all succeeding joints, there was measurable deformation. The solid metal of plate *C*, No. 25, and those aft displayed measurable longitudinal strains, due either to the vibrations of the engines or to the pitching of the ship or to their combined effect.

The magnitude of the strains in general increased as the bulkhead of the superstructure was approached, although the rate observed was not a uniformly increasing one. The presence of hatches probably influenced the magnitude and distribution of the stresses. The smooth sea was favorable for acquiring comparative results on different parts of the ship. At times the engines on one shaft would gain in the number of rotations over those of the other, and differences in strains were noted which were attributed to the effects of the heavier pistons when those of both shafts were in the same phase.

Strains across the lap joints of courses *C* and *D* were substantially the same and much greater than those of the stringer plates. This behavior calls for the development of longitudinal shearing stresses at the seams betwen the stringer course and the adjacent course, but measurements necessary to follow these shearing strains were not included in the present series.

Pitching strains were not of sufficient magnitude to admit of measurement on the deck amidship, covered by the superstructure, but those caused by the engines were followed to a point nearly abreast the boiler hatch.

On the after deck the strains reached a maximum near the bulkhead. The movement across one of the lap joints exceeded the scale reading of the gage, and was estimated

TABLE No. 2

STRAINS IN PLATES OF SHELTER DECK OF S. S. ANCON, WHEN AT SEA LOADED WITH 3,000 TONS CARGO. DRAFT, 16 FEET 4 INCHES FORWARD, 19 FEET 9 INCHES AFT

Measurements with scissors gage, lengths of 6 inches each, on deck plating and bulwark rails, port side of ship, on return trip Colon to New York, May, 1913. Observations on plates at middle of width and length, also stepping down from one plate to another across triple-riveted lap joints. Light sea. After part of ship.

Course of Plating	Plate		Gaged Length				Probable Effect of	
	No. of	Weight of, pounds	Location	On solid plate or joint	Measured Strain, Inches	Corresponding stress, pounds per square inch	Engines	Pitching
C	9 to 10	22	Just aft, after bulkhead of superstructure	Joint	Estimated 3 times scale reading			.00300 ±
C	8 to 9	22	Abreast forward end Hatch No. 6	"	.00075		.00005	.00070
C	8	22	" Hatch No. 6	Solid plate	.00022	1,100	.00002	.00020
C	7 to 8	22	" " " 6	Joint	.00070		00010	.00060
C	7	22	" " " 6	Solid plate	.00010	500		
C	6 to 7	22	" after end Hatch No. 6	Joint	.00050		Perceptible	.00050
C	6	22	Nearly abreast main mast	Solid plate	.00018	900	"	.00018
C	5 to 6	22	Abreast main mast	Joint	.00040		"	.00040
C	5	22	Just forward Hatch No. 7	Solid plate	.00012	600		
C	4 to 5	22	Abreast Hatch No. 7	Joint	.00020			
C	4	22	" " " 7	Solid plate	.00004	200		
B	3 to 4	22	Abreast after end Hatch No. 7	Joint	.00040			
D	8	16	Just aft, after bulkhead of superstructure	Solid plate	.00022	1,100	0	.00022
Bulwark rail	8 to 9	24	" " " " " " "	Joint	More than .0010)		.00015	.00085 ±
			Two feet aft after bulkhead of superstructure	Solid angle	.00045	2,250	.00005	.00040
				SMOOTH SEA.	AFTER PART OF SHIP.	STARBOARD SIDE.		
C	6 to 7	22	Abreast after end Hatch No. 6	Joint	.00040		.00005	.00035
C	6	22	Nearly abreast main mast*	Solid plate	.00012	600	0	.00012
C	4 to 5	22	Abreast Hatch No. 7	Joint	.00012		.00003	.00003
C	3 to 4	22	Aft, after end of Hatch No. 7	"	.00015		Perceptible	.00015
Bulwark rail	2	22	Near Bitts*		No strain detected			
			Two feet aft, after bulkhead of superstructure	Solid angle	.00040	2,000		
Bulwark rail			Forward of superstructure	" "	.00055	2,750		

* Vertical vibrations in deck very pronounced at these places.
In vicinity of main mast very pronounced vertical vibrations, but no strains detected.

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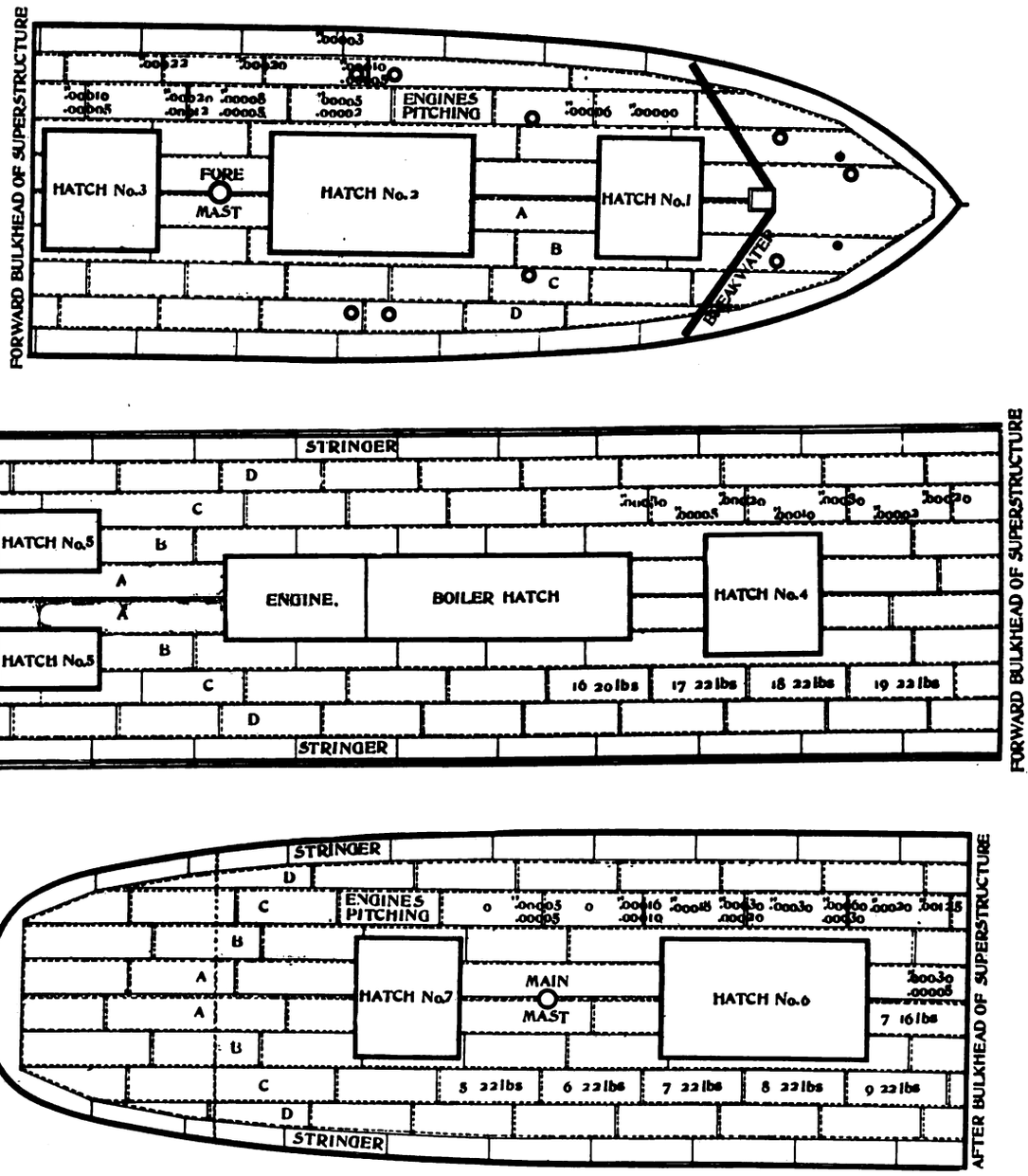
to be .00125 ±. Strains due to the engines were greater than those due to the pitching of the ship, both forward and aft the superstructure. This remark concerning the relative effect of the engines and the pitching of the ship refers to the observations made on the outward voyage. Northbound, with a light cargo, the relative effects of the engines and the pitching were reversed over those of the outward voyage, the pitching strains now predominating.

Strains due to pitching and those due to the engines were recognized by their periodicity. The engines made 78 revolutions per minute and strains attributed to them were developed synchronous with the revolutions.

In explanation of the relatively large movements due to the pitching of the ship when in comparatively smooth water, it appears probable that the lap joints of the deck plates had become loosened in a degree by exposure to repeated stresses.



Figs. 5, 6 and 7.—Some of the Measured Strains, Effects of Engines and Pitching of Ship, Smooth Sea. Gaged Lengths, 6 Inches. Full cargo

A comparison of results of the outward and return voyages is shown on Fig. 8. Whereas on the outward voyage a strain caused by the engines of about .00125-inch was observed at the lap joint of course *C*, next the bulkhead; on the return trip the effect of the engines was hardly perceptible at that place, but the pitching of the ship caused a movement estimated at three times the scale reading, or .003 ± inch. At other joints and on the solid plates similar results were observed, the pitching strains greatly predominating on the return voyage. All measurements were taken along the middle of the width of the several courses of plates.

A portion of the movement observed not unlikely represented slip of the joints. Of the effects which came from the engines there would be some seven hundred thousand to eight hundred thousand repetitions during one voyage to the Isthmus.

Places on the deck where the greatest longitudinal strains occurred were not usually accompanied with marked vertical vibrations. At least certain places where vertical vibrations were most pronounced, under the feet, were examined without finding unusual longitudinal strains. On the deck over the ridge bar on the starboard side, and also near the

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main mast were places of pronounced vertical vibrations, but where the measured strains were of moderate extent. Again, at the stern, near the bitts, the vertical vibrations were very marked, while the longitudinal strains in the vicinity were not measurable.

MAXIMUM STRESSES MEASURED

The stresses corresponding to the measured strains, observed on the forward deck, ranged from 100 pounds to 650 pounds per square inch. On the after deck, 2,250 pounds per square inch was found, all referring to measurements made on a smooth sea. The stresses reached a maximum in the vicinity of the bulkheads of the superstructure, and disappeared as the bow or the stern was approached.

The observations which were made, prior to the main series, when a moderate sea was running, showed stresses on the after deck near the bulkhead ranging from 2,500 to 3,500 pounds per square inch. The highest stress observed was on the bulwark rail, where a stress of 5,500 pounds per square inch was found. Fig. 9 shows the place occupied by the strain gage on the angle of the rail when this observation was made.

The measured strains represent changes in length of the plates, and it was obvious at the time of their development which they were, whether they were in a tensile or a compressive direction as the gage successively opened and closed, but pre-existing strains in a finished structure at repose defy definition. In order to ascertain what strains exist in different parts of the ship, of tension and compression, or zero strains at neutral zones, it is necessary to establish reference lengths initially on the component members and remeasure them at desired stages. It involves considerable work to acquire reliable information upon the distribution of strains in the principal members of a ship, when launched, with the machinery installed, with a cargo aboard, and finally the live loads at sea, but it is entirely feasible to acquire such data, and in fairly complete detail. The method of measured strains devised by the writer, and which has been used in the investigation of internal strains in steel forgings since 1887, and more recently in the investigation of strains in engineering structures, admits of furnishing this information. This method of investigating strains was employed on the present occasion in measuring the deck plates during the time the ship was receiving cargo.

STRAINS RESULTING FROM LOADING CARGO AND FROM CHANGES IN TEMPERATURE

The purpose of making the strain gage measurements at the time of taking cargo aboard was primarily to ascertain what strains resulted directly from loading the ship. Notwithstanding the intent of the observations, the results evidently showed effects which were chiefly due to changes in the temperature of the plates, with obscure effects, if any, attributable to the cargo. With stevedore work going on at the several hatches few places on the deck were accessible at all times, or available even in the early hours of the morning, when most of the measurements were made. The places selected were just aft the bulkhead, as shown by Fig. 10.

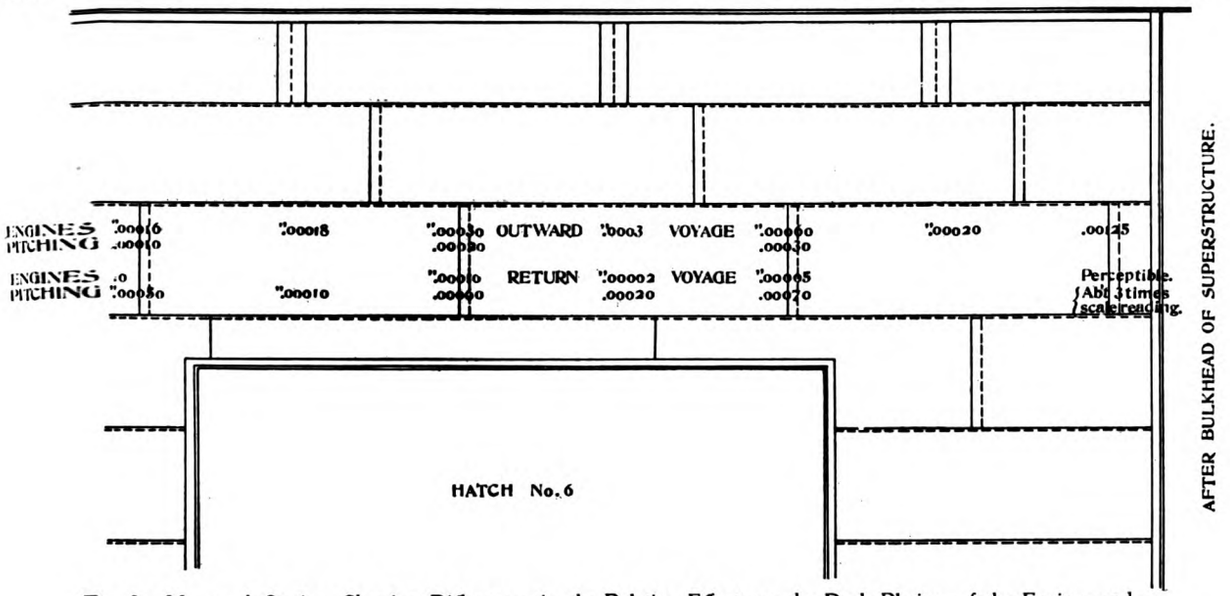

Fig. 8.—Measured Strains, Showing Differences in the Relative Effects on the Deck Plating, of the Engines and the Pitching of the Ship, According to Amount of Cargo Carried. Outward Voyage Made with 10,217 Tons Cargo; Return Voyage Made with 3,000 Tons Cargo

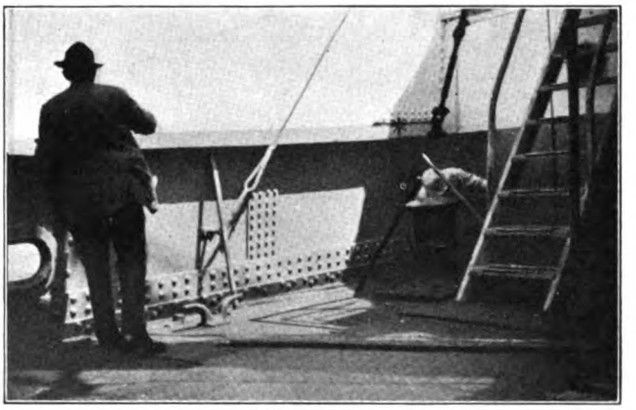

Fig. 9.—Place, Marked *, on Bulwark Rail where Maximum Strains were Observed

Reference lengths were established, of 20 inches each, on six courses of plating, three on each side of the ship, and one on each bulwark rail. The latter were on the angles directly over the double-riveted butt strap joints of the bulwarks. All reference lengths were fore and aft in their direction.

Measurements were inaugurated soon after the ship came out of dry dock, and were continued until the day of sailing. Fig. 11 shows a diagram upon which are entered the results of these measurements, also the temperature of the deck at each of the places measured, the hour of the day, and the draft of the ship forward and aft.

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On June 13 the ship was at dock in Brooklyn, with bow heading East; on June 15 and 16 at Communipaw with bow toward the West. On succeeding days at Pier No. 52, North River, with the bow heading East. These changes in position introduced differences in thermal conditions which doubtless influenced certain of the measurements.

On the morning of June 20, sailing day, the temperature of the deck was nearly uniform, the range being from 64 to 67 degrees F. For convenience, the stresses were regarded as having zero value at that time, and the state of strain or strains on other days is given with reference to those on the day of sailing.

The manner in which the strain gage readings were taken and results arrived at were as follows: A reading is taken with the strain gage on the reference bar, followed by a reading with the gage on the deck plate. The distance or gaged length laid off on the deck is thus determined with reference to that laid off on the reference bar, the strain gage being used as a transfer instrument to compare one gaged length with another. The reference bar is assumed to have a length of exactly 20 inches. The temperature of the reference bar and that of the deck is noted. If they are not the same the reading for the deck length is corrected to correspond with that of the reference bar. The latter is made of structural steel or steel of that grade and assumed to have the same coefficient of expansion as the deck plates.

Assuming the strains to have zero value on the deck plates on some given day for comparison, on this occasion taking the lengths as they were measured on June 20, the individual lengths are said to be in tension or compression on other occasions according to their corrected lengths, that is, according to whether they were longer or shorter than on the day selected for comparison. Referring to examples on the diagram Fig. 11, the corrected gaged length of stringer plate No. 10, port side, on June 13 was .0019-inch shorter than it was on the 20th. The force required to shorten the plate this amount is 2,850 pounds per square inch compression. On June 16 the same gaged length, corrected, measured the same as on the day selected for comparison. It therefore had zero stress upon it with reference to its state on June 20. Again the gaged length was .0008-inch longer on the 18th, than on the day selected for comparison, which strain corresponds to a stress of 1,200 pounds per square inch tension.

The results exhibited on the diagram Fig. 11 were deduced in the maner described. Where stresses of tension are shown on the diagram the corrected gaged length on the deck was longer than it was found on the day selected for comparison, namely, June 20. Where stresses of compression are shown the gaged length was correspondingly shorter.

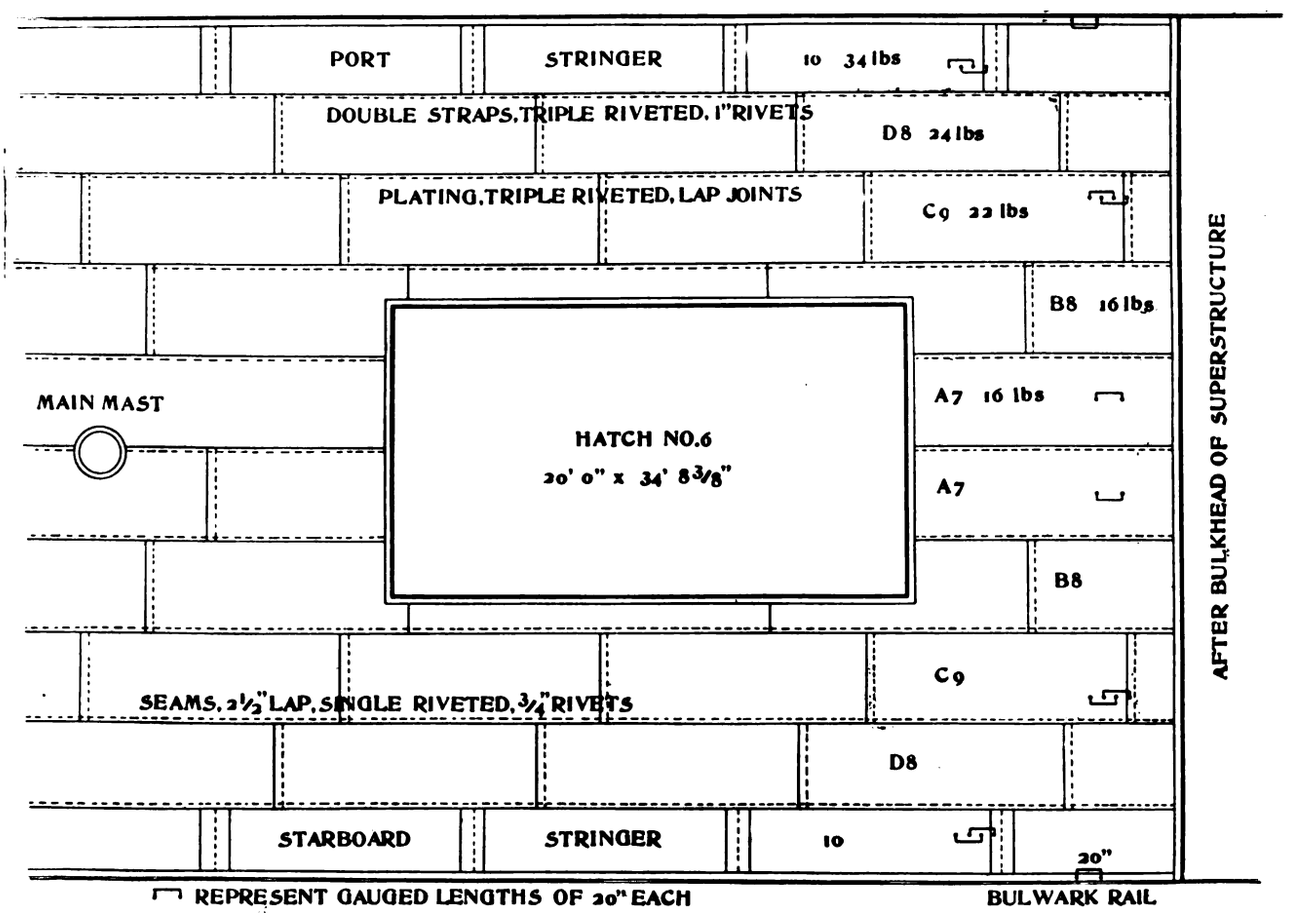

Fig. 10.—Strain Gage Measurements in Plates of Shelter Deck, Made While Receiving Cargo

There was no change in the cargo between the time of measurements of June 15 and 16, but a wide difference in temperature, accompanied by a wide difference in the stresses. The starboard stringer plate passed through a range from 2,550 pounds per square inch compression to 1,050 pounds per square inch tension, a total of 3,600 pounds per square inch. This and other changes which occurred between those dates were clearly temperature effects.

Significance is attached to the similarity between the shaded parts of the diagram representing the stresses in the stringer plates and bulwark rails on June 13 and June 15, comparing the port side on the first day with the starboard side on the second day. The ship was moved between those dates from

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Brooklyn to Communipaw, and turned half around. Both days were sunny, and it would seem that the effect of the sun on the port side when at Brooklyn was closely reproduced on the starboard side when at Communipaw.

The state of the strains on the mornings of June 16, 19 and 20 were nearly the same. In the meantime, cargo was received which increased the draft from 12 feet 6 inches to 26 feet 6 inches forward, and 19 feet 6 inches to 27 feet 6 inches aft. It is not improbable that some effect resulted from taking this amount of cargo aboard and bringing the ship nearly to an even keel, but the part belonging to temperature effects and that belonging to the cargo does not admit of being separated.

in the lap joints of courses *C* than in the solid plates. Dotted lines on the diagram connect plotted points showing the movements at the joints, located according to the same scale as the plotted strains in the solid plates.

It will be noted, for example, that the changes in strains from June 13 to the 15th on the port side was much greater for the gaged length spanning the lap joint of course *C* than the strains in the solid plate, more than twice as much, while on the stringer course the reverse was true. Here again the general looseness of the lap joints appears in evidence.

Complex thermal conditions prevailed throughout this series of observations, but no more than would be experienced on

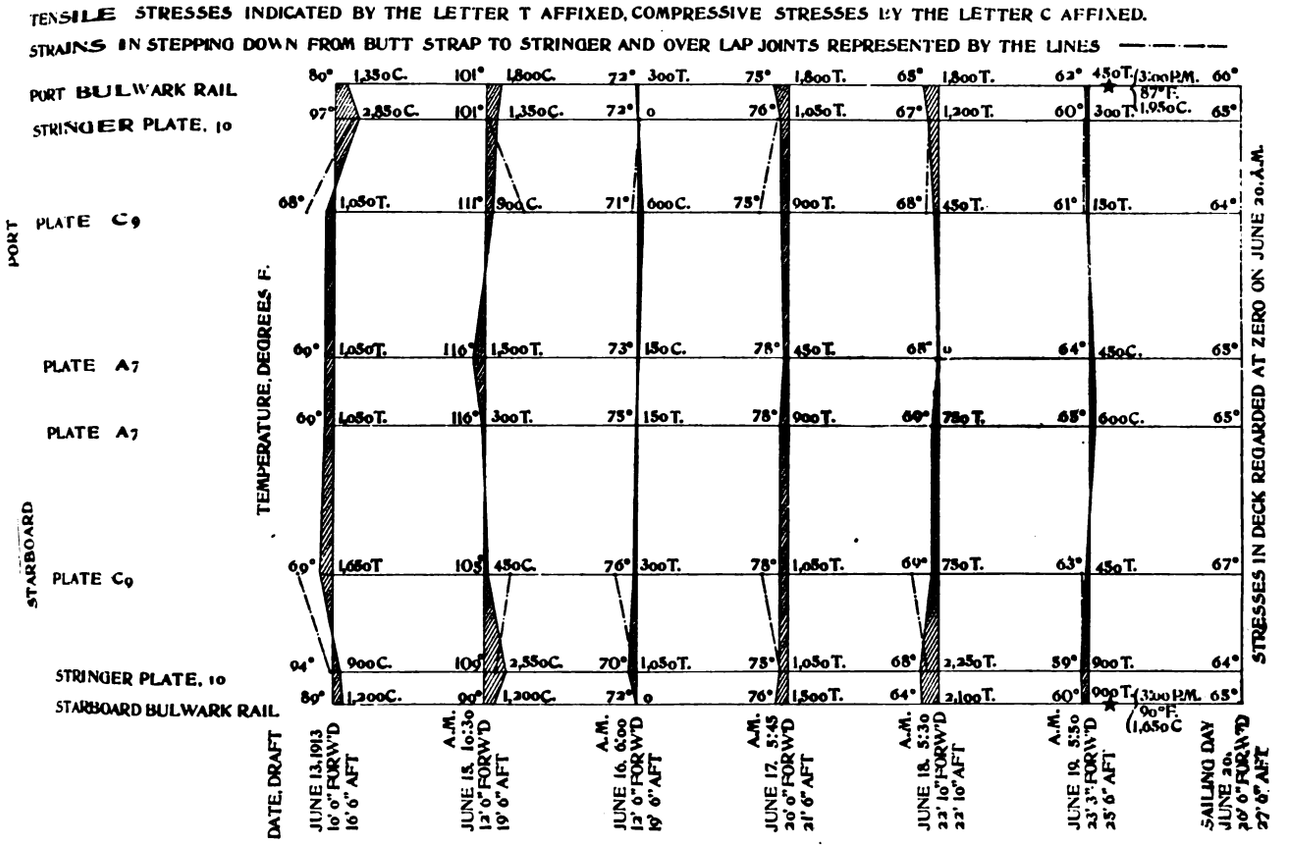
TENSILE STRESSES INDICATED BY THE LETTER T AFFIXED, COMPRESSIVE STRESSES BY THE LETTER C AFFIXED.
STRAINS IN STEPPING DOWN FROM BUTT STRAP TO STRINGER AND OVER LAP JOINTS REPRESENTED BY THE LINES
PORT BULWARK RAIL
STRINGER PLATE, 10
PORT
PLATE C9
PLATE A7
PLATE A7
STARBOARD
PLATE C9
STRINGER PLATE, 10
STARBOARD BULWARK RAIL
TEMPERATURE, DEGREES F.
STRESSES IN DECK REGARDED AT ZERO ON JUNE 20, A.M.
DATE, DRAFT
JUNE 13, 1913 10' 0" FORW'D 16' 6" AFT
A.M. JUNE 15, 10:30 12' 0" FORW'D 19' 0" AFT
A.M. JUNE 16, 6:00 12' 0" FORW'D 19' 0" AFT
A.M. JUNE 17, 5:45 20' 0" FORW'D 21' 6" AFT
A.M. JUNE 18, 5:30 22' 10" FORW'D 22' 10" AFT
A.M. JUNE 19, 5:30 23' 3" FORW'D 25' 6" AFT
SAILING DAY JUNE 20, 20' 0" FORW'D 27' 0" AFT

Fig. 11.—Diagram Showing Stresses, Pounds per Square Inch, Corresponding to Strains Measured on Plates of Shelter Decks While Ship Was Receiving Cargo

On June 20 the usual complement of readings was taken in the morning, supplemented by observations on the bulwark rails during the day, the temperature in the meantime having risen from 60 degrees to 90 degrees on the starboard side and 62 degrees to 87 degrees on the port side.

On the starboard side the stress changed from 900 pounds per square inch tension to 1,650 pounds per square inch compression, and on the port side from 450 pounds per square inch tension to 1,950 pounds per square inch compression. These changes again clearly represent temperature effects.

Higher stresses were developed in the more rigid stringer plates and the bulwark rails than in the plates having lap joints, a result to be expected whether caused by temperature changes or by taking aboard the cargo.

The maximum range in stresses observed were 4,800 pounds per square inch on starboard stringer plate No. 10 and 4,050 pounds per square inch on the corresponding port stringer plate. The range in temperature for all places observed was from 60 deprees to 116 degrees F.

Four of the gaged lengths spanned joints in the plating. There was greater apparent rigidity in the butt joints of the stringer courses than in the solid plates, and less rigidity other occasions. The ship floated in water at about 60 degrees, setting deeper and deeper as the cargo was received, while the temperature of the deck plates reached 116 degrees on one occasion.

In conclusion, attention is invited to the opportunities afforded by engineering structures of all classes in the acquisition of information upon the behavior of materials in service, the distribution of stresses, and verification of computed stresses, along the lines of inquiry illustrated in these strain gage observations.

SHIPBUILDING RETURNS.—The Bureau of Navigation reports 101 sailing, steam and unrigged vessels of 15,409 gross tons built in the United States and officially numbered during the month of March. Six of these were steel steamships aggregating 6,435 gross tons. During the nine months ending March 31 eight hundred and eighty-one vessels aggregating 215,059 gross tons were built, of which sixty-nine aggregating 124,275 gross tons were steel steamships. The construction during the corresponding period a year ago amounted to 1,114 vessels of 300,265 gross tons.

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Some Questions Relating to Battleship Design*

BY T. G. OWENS

Many and conflicting are the views which are held and have been put forward by experts as to what constitutes the most desirable battleship—*i. e.*, that which embodies the most commendable compromise of conflicting elements to suit the requirements of strategists and tacticians. It may therefore be of interest to outline in one paper the most noteworthy of these views, and to see whether it is at all possible, out of such contradictory elements, to evolve some one proposal which could reasonably be said to embody the desiderata and to eliminate the objectionable features of a battleship with respect to armament, armored protection, speed, dimensions, etc. Such consideration of the subject is perhaps justified, since one feature of the modern line-of-battle ship frequently criticised is the ever-increasing displacement, the normal displacement of some of the most recent types being over 50 percent in excess of that of the Dreadnought of some eight years ago.

Principal Features of Modern Design

The present tendency in warship construction and design, as exemplified in the later ships of all the principal maritime powers, is towards a very large displacement, with the arrangement of all the guns of the primary armament on the center line and with the guns of the auxiliary or secondary armament placed in an armored citadel on, or immediately below, the deck carrying the primary guns; while in respect to the above-water armored protection there is the usual thick armored belt, extending, say, from 4 feet to 6 feet below the waterline to the height of the main deck, and extending over the length of the vessel for such distance as to protect the machinery and boiler compartments, and also the magazines. The armor protecting the primary armament corresponds in thickness in most cases to that of the main waterline belt previously alluded to. The ends of the ship and the citadel or battery for the secondary armament have armor of reduced thickness, approximately half that of the main belt protection. In all the modern battleships there are horizontal protective decks, and in most of the later ships there are also submerged longitudinal protective bulkheads to ward against the effect of submarine explosions, either from torpedoes or mines.

The arrangement of the guns and the distribution of the various thicknesses of armor differ practically with each class of vessels building for each country; but all more or less follow the same general lines, and are designed for the same purposes. It would appear as if matters would go on in this way until either the naval authorities throughout the world convince themselves, or the result of a naval engagement shows, that the thicknesses and the extent of armor as at present fitted are inadequate to afford the protection desired or necessary, or that a reduction in the armored area is permissible without endangering the battle qualities of the vessel; or, on the other hand, that the guns of the primary armaments are larger and more powerful than are called for by the protection given, or that the auxiliary or secondary armament is useless, and could, with its protection, be removed without detriment to the fighting qualities of the vessel. Any one of the above cases, or a combination of them, would, of course, lead to a very considerable reduction in the displacement of vessels, always supposing that the number of guns remained approximately constant, and that the speeds now obtained are considered adequate.

The late Sir William White, in 1911, in a paper to the American Society of Naval Architects and Marine Engineers, stated that the limit of reasonable maximum dimensions had been reached, and that a decrease in the number of primary guns and a retention of the same speed would considerably reduce the size and cost of individual ships. The reduction in the number of primary guns has already taken place, at least in Britain and Germany, but the displacement and cost continue to grow.

It is not easy to formulate a design which one can put forward with confidence as, what has been termed, the most desirable design in every way; but it appears to the author that it is quite possible and reasonable to outline certain conditions which must be fulfilled in order to obtain a ship-of-the-line capable of holding her own in any action whereever and however fought, and to combine such qualities in a vessel whose displacement would, as present displacements go, be moderate, any increase being solely due to academic or departmental insistance on what may be called "trimmings," not at all necessary to the fighting qualities of the vessel.

Elements of the Most Desirable Design

Primary Guns—Taking first the gun and torpedo armament, it will be agreed that the primary gun and projectile should be of the smallest caliber and weight necessary to carry an efficient bursting charge through the maximum thickness of protecting side armor on a ship of a possible enemy at a range of maximum vision; it should at the same time possess the greatest possible length of danger zone without unduly affecting the life of the gun. Giving all these features due consideration, there appears no great necessity in the present or in the immediate future for guns of larger caliber than 14-inch, which at the range of probable maximum vision, say 12,000 yards, would, according to the Gavre formula, and allowing for angle of descent, pierce the best Krupp armor of a thickness of 14 inches and deliver a heavy bursting charge into the interior of the ship. But guns of larger caliber than 14 inches are already mounted, and a further increase is not outside the bounds of probability.

Arrangement of Primary Guns—As to the arrangement for mounting the guns of the primary armament, it will probably not be disputed that a four-turret center-line arrangement, if carrying sufficient gun power, is, on the score of supervision, smaller target, and smaller number of gun crew required for working, preferable to that of a five or six-turret arrangement; and if such turrets, with their magazines, are separated from each other by a considerable length of ship, the chances are lessened of simultaneous destruction of two complete turrets and magazines by one lucky shot, which, under present conditions, is quite possible.

It may, perhaps, also be agreed that, if all primary guns were arranged to have their axes approximately at the same height above the waterline, the work of those responsible for getting in the largest number of hits in the shortest possible time would be immensely facilitated. This would mean the adoption of a flush-decked ship, the end-on fire would consist of that from the guns of one turret only, which, as end-on fire is now at a discount, would not detract from the value of the ship. The high forecastle would go, the depth molded of the vessel would be increased to give an ample freeboard, and the main deck would then be of sufficient height above the waterline to become the gun-deck for the auxiliary armament.

Auxiliary Armament—As regards auxiliary armament, if secondary guns are to be carried at all, it will be conceded that these should be of as large caliber as can be conveniently man-handled in respect, not only of charging the gun, but in training and elevating it with considerable rapidity. They should be mounted so as to give the highest possible maximum of fire forward and aft, and to possess the largest arcs

*Extracts from a paper read before the Institution of Naval Architects, London, April, 1914.

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of training consistent with adequate protection. The height of their axes above the waterline should be such as to enable them to fight in a reasonable seaway, but at the same time this should not be exaggerated, as undue height leads to a large expenditure in weight of structure and armor protection, and to increased target. They should also be as numerous as the conditions of displacement and accommodation will allow.

This arrangement of main and auxiliary armament in its main features accords more or less with that proposed by the late Sir W. H. White in 1910; he wished it placed on record that "in no case is it desirable to carry more than eight heavy guns in a single ship, that these guns are best arranged in four positions, and that they should be supplemented by a powerful and well-protected secondary armament."

Mr. Alan H. Burgoyne, M. P., in his paper read at the last spring meeting of this Institution, suggested that the reversion of the British Admiralty to an anti-torpedo craft armament of 6-inch guns, in their latest ships, was a tacit, if belated, recognition of an error in judgment; certainly no other maritime power followed their lead in dispensing with the secondary guns, least of all those who bore the brunt of the last great naval war, and who knew the merits and demérits of the arrangements placed under their control.

Torpedo-Tubes—The new large-type torpedoes of extraordinary range possess a real as well as moral worth, and in order to utilize such weapons it appears desirable to fit as many broadside submerged tubes as space can be found for within the limits of the dimensions of the ship, and preferably not less than three on each broadside.

Armor Protection—The conclusions arrived at in the preceding part of this paper accord with actual experience and with information gained from reliable sources as to experiments carried out in many countries, and induce the author to put forward as vital:

1. That the waterline area should be protected with armor from a reasonable depth below to a reasonable height above the normal waterline, preferably to the main deck and for the whole length of the vessel, that portion extending along the machinery spaces and magazines to be of such thickness as would probably prevent penetration by projectiles fired from a gun of 14-inch or 15-inch caliber at a range of 10,000 yards, and having the highest practicable initial velocity.

2. That the fixed armor of the barbettes or turrets should be of a thickness not less than that of the waterline belt.

3. That the fronts of the revolving gun-shields for the primary armament should be at least 10 percent thicker than the thickness of the waterline belt.

4. That the conning tower should be protected by armor 20 percent thicker than that of the waterline belt; and

5. That the thickness of the armor protection of the secondary armament should be at least capable of excluding armor-piercing projectiles from the largest man-handled guns of a possible adversary at the decisive fighting range of 8,000 yards.

The late General Cuniberti expressed the opinion in 1912 that battleships should be protected with armor at least 12 inches thick, and, recognizing the practical impossibility of covering the whole side, he suggested that it should more particularly apply to those parts which it is vitally necessary to keep intact until the end of an action.

As to the extent and thickness of armor for the remaining portion of the ship, the main waterline belt should be continued almost to the ends with armor of the same thickness as that protecting the secondary armament, and thence to the ends with armor of not less than 4 inches. Forward of the citadel the vessel should be protected to the height of the main deck, but it was not necessary to carry the armor so high at the after end. The need for increased height of armored end protection forward was established in the last great naval battle; the unarmored forward ends of some of the vessels were completely riddled by quick-firing guns, with the result that the protruding and jagged shell plates coupled with the inrush of water as the vessels tried to steam ahead made the ships practically unmanageable.

Under-Water Protection—The longitudinal protective bulkheads below the waterline protective deck should, if experiments count for anything, be at least 15 feet from the ship's side, and should extend as far as practicable throughout the length of the ship. The construction should be such that that part of the protective deck between the top of such bulkheads and the outer plating will be subject to fracture before the longitudinal protective bulkheads are stressed up to their limits by the confined gases in the outer wing compartments, due to any submarine explosion, but it is difficult to arrange for this in a reliable manner. The distance of the bulkhead from the side of the ship appears to be of more importance than its actual thickness. One experiment carried out showed that a thick bulkhead fitted at the usual distance from the side was broken up by the force of a submarine explosion, and was hurled through the inner bulkheads, destroying the boilers in its course, and more damage resulted apparently than if such bulkhead had not existed.

Proposals have been made to fit under-water broadside protection in such a way as to provide a more or less elastic shield, which would to some extent minimize by its flexibility the effect of the full force of the explosion. Some time ago the author's staff worked out and elaborated a suggestion of Commander Elia for this purpose, and although the arrangement is somewhat expensive, it is apparently more capable of fulfilling the purpose for which it is designed than any other yet proposed.

Internal Subdivision—The question of internal subdivision also assumes a large and important place. The more the internal space both above and below the waterline is subdivided into watertight compartments, the less likelihood is there of the vessel being put out of action either by shell fire or submarine explosion. The views of Professor Hovgaard, recorded in the Transactions of the American Society of Naval Architects and Marine Engineers of 1903, furnish an eminently practical standard for such internal subdivision.

It should also be borne in mind that the same aggregate amount of damage would occur to a small as to a large ship from torpedo attack, and it will readily be seen that an explosion which might possibly put the small ship out of action might leave the large ship capable of maneuvering and continuing in action. As an example of what it is possible to do in the way of meeting such under-water attacks, reference may be made to a foreign warship, designed by the author, and built by Vickers under a contract stipulation that, in the event of 50 feet of the side being blown away amidships, the vessel should not heel over more than 11 degrees, and automatically regain the vertical in a certain specified time, although necessarily with an increased draft due to the buoyancy lost.

Horizontal Protection—Whether the vertical side armor extends to the height of the main or to the upper deck, it is very necessary that horizontal protection of adequate thickness and quality should completely cover it like a lid, extending at least for the full length of the gun citadel, and from armored side to armored side. In the case where the height of the armored side is lower at the ends, the level of the horizontal protection should coincide with it. It is a matter of difficulty to determine with anything like exactitude the correct thickness for such protection. Experiments carried out with plunging shot, with a view to establish some guide in the matter, indicated that it should not be less than 1½ inches of mild steel, or, preferably, of higher quality steel,

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possessing the necessary characteristics to act as part of the structure of the vessel, and not purely as horizontal armor involving additional weight—an arrangement sometimes to be found in the ships of foreign navies.

Protective Decks—The above is outside the question of the protective decks proper, either angled or horizontal, in the region of the waterline, or any intermediate deck which may be used for the purpose of protection. The author is rather skeptical as to the value of the angled protective deck, believing that its enormous weight, where it adds least to the structural strength, would be much better employed in thickening the vertical side armor or increasing the horizontal deck forming the armored lid to the side armor. The aim should be to keep out the explosive shell rather than to arrange for its reception in the interior of the ship. The protection afforded by such deck is more apparent than real; but as it will be difficult to convince the men who work below of its weakness, it will probably continue to form a feature of our vessels.

With respect to the value of protective decks, Professor Hovgaard, in his paper of 1903, stated that, while they probably afford the best protection for a given weight against armor-piercing shots, they are far less effective against bursting shells, and that when a high explosive shell bursts against a sloping deck it always results in a great breach. There is every reason to believe that this opinion is correct.

Protection Against Air-Craft—Protection against attack from bombs, etc., dropped from air-craft is not yet in the region of practical politics, and the seaman does not appear as yet to be worried by such danger. When the time arrives to arrange measures to meet such attack they will probably take form in the method advocated by Sir Trevor Dawson—*i. e.*, to increase the thickness, and give a greater curvature, or whale-back formation, to the armored lid before referred to.

Machinery—The principal feature which distinguishes the propelling machinery of battleships of the Dreadnought era from that of an earlier date is the adoption of steam turbines. By their introduction the great advantages of an increase in power per unit of weight was immediately obtained. Thus the new type of machinery simplified the problem of providing for higher speeds with increased displacement due to the armament and protective material. In pre-Dreadnought battleships, powers reaching approximately to about 18,000 indicated horsepower were obtained with a weight of about 1,760 tons, which is equal to 10.2 horsepower per ton of machinery with steam up, whereas in the earliest turbine-driven battleships the power of the machinery was increased to 25,000 shaft horsepower, and 12¼ to 13¾ shaft horsepower was easily obtained per ton of machinery weight.

Twin screws working at comparatively low revolutions gave place to quadruple screws running at a greatly increased speed. Although unfavorable to propeller efficiency, yet the combined efficiency of turbines and propellers, as compared with reciprocating engines and propellers, was such that the former was decidedly favorable to reduction in weight and increase in steam economy. The new type of machinery, being more economical at the higher powers than the reciprocating engines, enabled a reduction to be made in the proportions of the boilers, with a corresponding reduction of weight and of the space occupied.

Respecting radius of action at low speeds, the performance of the earlier types of turbine-driven battleships was somewhat disappointing, but with each successive type improvement was made in this respect. In the earliest ships cruising turbines were installed to drive direct the shafts worked by the main turbines. Later, the cruising turbines were dispensed with and a cruising element was introduced at the forward end of the high-pressure turbine. In the latest designs a separate high-speed turbine driving the high-pressure or low-pressure main turbine line of shafting through gearing seems to find favor and promises most favorably as regards economy at low powers. The fitting of cruising turbines, either geared or direct-driven, involves an increase in the length of the engine room, but the gain which results from the greater economy perhaps justifies this.

As regards type of boiler, there is little to choose between the two generally adopted in British battleships—viz., Babcock and Wilcox and the three-drum straight tube or Yarrow type. The same duty can usually be obtained from either on a given weight or for a given space for a battleship installation. The heating surface, however, is somewhat less for the Babcock and Wilcox type, but in the Yarrow boiler the fire-grate area is smaller. Both appear to be suitable for oil fuel as well as coal, and coal and oil combined can be burned with equal facility, if required, in both.

The exclusive use of oil fuel, as proposed for some recent battleships, effects considerable saving in weight and space for a given power. Moreover, the adoption of oil fuel to the exclusion of coal simplifies the bunkering of the ship, reduces the number of stokers, and enables steam to be maintained with greater ease and for a longer duration of time without cleaning boiler tubes and uptakes. Indeed, oil fuel is altogether desirable from the naval engineer's and naval architect's standpoint. Although the advantages in reduced displacement, higher power, and speed on a given weight, lies entirely in favor of the purely oil-consuming boilers, there is no doubt, in the author's opinion, that, under present conditions as to sources and storage of coal and oil supplies, the safer and wiser policy lies in a combined installation. This would enable a vessel more readily to obtain a constant supply of fuel in war time than would be the case were oil the fuel exclusively used.

The internal combustion engine, although its design and manufacture has rapidly progressed in recent years, has not yet been applied to the propulsion of capital ships. In submarines and in some vessels of the mercantile marine it has undoubtedly proved a great success. The development of the internal combustion engine has now reached a stage at which some engineers are prepared to install these engines in battleships of moderate power and speed. The possibility of failure, with the enormous expense involved, has so far deterred the responsible authorities of maritime powers from arranging for such an installation in large ships, so that for the moment in considering the type ship, such machinery must be omitted. Even in cases where it might be eventually fitted, we are face to face with the fact that, notwithstanding the possible advantages in weight and space to be derived from its adoption, and the undoubted greatly increased radius of action, it would mean a return to the old principle of the reciprocating type with its necessity for balanced movement and compensations, which does not exist with the present system of turbine installations.

Many look with confidence towards utilizing the rotary motion of turbines with some necessary modifications, while dispensing with the present boiler arrangement, by an installation arranged to produce an expansive motive fluid by means of the combination of the products of combustion of small internal combustion units with compressed heated air, such motive fluid being sufficiently cool in order to obviate injury to the turbines. Such an arrangement has already been designed and apparatus of a comparatively large power constructed, on which confidential experiments have been carried out. These point in the direction of ultimate success. Such an installation, while retaining all the advantages of internal combustion machinery with respect to economy of fuel consumption, would at the same time have all the immense advantages of the turbine with its balanced action and unlimited capacity for adapting itself to varying loads. Such

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a turbine would not in any sense correspond to what is usually termed a "gas" turbine, nor would it suffer from the drawbacks attendant upon the immense heat which gas turbines are called upon, but fail, to stand. The matter is mentioned here incidentally to show that, even accepting internal combustion engines of sizes suitable for battleship propulsion to be an accomplished fact, the last word has not yet been said.

Finally, in connection with machinery arrangements, the flexibility of the structural portion of long vessels requires some consideration. It apppears very necessary to introduce some device in the form of a flexible coupling in a position between the engine room and the shaft tube, so that the deflections of the ship among waves shall have the minimum effect on the alinement of the shafts.

DESIGNS OF THE "MOST DESIRABLE SHIP"

Based on the foregoing conclusions, which go to make up what has been termed "the most desirable ship," there is evolved:

1. Design I, which shows eight 14-inch 45-caliber guns, twin mounted, arranged in widely-spaced barbettes, with the axes of the guns at the same height above the waterline in each case—viz., 27 feet 6 inches.

2. Design II, showing an alternative arrangement, in which eight 14-inch 45-caliber guns are twin-mounted in barbettes. These barbettes, however, are arranged in pairs at each end of the ship, with the axes of the guns placed at different heights above the waterline.

The principal particulars of both ships, in addition to those already stated, are:

Dimensions—580 feet by 91 feet by 27 feet 6 inches draft.

Speed—21½ knots for a continuous trial of 12 hours' duration, burning coal and oil.

Displacement—25,500 tons in the normal condition.

Secondary Armament—Fourteen 6-inch quick-firing guns.

Submerged Torpedo Tubes—Six 21-inch broadside.

Armor—The main waterline belt is of 13-inch K.C. armor, reduced to 6-inch and 4-inch at the ends. The fixed armor of the barbettes, where unprotected by other armor, is of K.C. quality, and equal in thickness to the main waterline belt. The front portion of the gun-shields is 14 inches in thickness. The secondary armament is in each case protected by K.C. armor not less than 6 inches thick. The conning-tower armor is 16 inches thick, of K.C. quality. Below the main waterline belt 3-inch K.C. armor 30 inches deep has been worked.

Machinery—Parsons' geared turbines on four shafts, with cruising turbines on each shaft.

Boilers—The boilers are of large watertube type, arranged for moderately high air pressure.

It will be seen that the vessel shown by Design I is in every way equal to the vessel shown in Design II with the exception of the end-on fire, and, in addition, has the various advantages already stated as to distribution of turrets; but, as custom is very strong, it is probable that the arrangement shown in Design II will more or less continue to prevail. It will probably be contended that the features outlined in Designs I and II exist more or less in present ships either built or building; in such case this is, in the author's opinion, so much the better for the ships, seeing that the deductions in the paper have been made not directly from what actually exists in present ships, but from what, if the arguments are correct, should exist to meet present conditions. The everyday experience of a warship designer brings him so much in contact with foreign authorities in discussing characteristics and details of warship construction that the author cannot conclude a paper like this without bearing testimony to the immense amount of skill, foresight, and common sense brought to bear in the evolving of designs, or in drafting the conditions to be met by designers, and to the real extent of which progress in warship design, as a whole, is furthered by experts of other countries than England. As a passing example, it may be mentioned that at the time when the main-drainage arrangements in British ships consisted of one or more continuous longitudinal drains practically connecting up each large compartment in the ship, many foreign vessels had each main watertight compartment quite independent in such respect. The same applies to the method of ventilation, both of which systems have since been adopted in this country. This point is merely given to show that, notwithstanding the great extent to which all maritime countries benefit by every advance made in British naval construction, we also receive valuable suggestions from those countries for whom we build, and also from those who build exclusively for themselves.

The author desires, in concluding, to express thanks to Sir Trevor Dawson for some of the later information respecting artillery; to Mr. McKechnie, in connection with the part referring to steam machinery; and to the author's staff for their assistance generally. He further wishes to acknowledge his obligation to various authorities and sources, whether mentioned in the paper or not, for much valuable information.

The Rational Non-Diesel Marine Oil Engine

BY ALBERT H. ZIEGLER

Although America may be justly proud of the high degree of engineering skill and the effective manufacturing methods displayed in the production of very large gasoline (petrol) marine engines, nevertheless it must be admitted that in the production of large marine oil engines America has long ago been fairly outstripped by most of the European countries. The internal combustion marine engine industry is realizing the fact that further material development and important extension of the field of application for its product, especially in the larger deep sea-going craft in competition with modern steam machinery, is not to be hoped for until the industry has settled definitely upon some one all round commercially practical system of fuel utilization which will promote the successful combustion of the very heaviest and cheapest of residue and refuse fuel oils instead of gasoline (petrol).

The oil engine itself is of course by no means a new subject for investigation and experiment to American manufacturers, many of whom have spent considerable money in making very earnest and consistent attempts at the solution of its problems ever since the very outset of the internal combustion engine industry. Until recently, however, not only have the ultimate attainments in American oil engine development been in no wise particularly notable or successful, but in only two or three instances where American builders have bought the patent rights from European interests have their products proved even moderate commercial successes.

The main reason for this has been that nearly all American manufacturers have for conservative business reasons refused to accept the Diesel oil engine as the only way out. They have chosen to experiment with slightly modified units of their regularly manufactured and marketed gasoline (petrol) burning engines. Indeed, they have shown an unmistakable gasoline (petrol) engine bias and have sacrificed too readily some of the most important theoretical functions of correct internal combustion engine design for the sake of obtaining some detail result of minor importance.

The non-Diesel oil engine may be defined as an internal combustion engine which takes oil fuel into the working cylinder during the suction stroke along with the air charge, atomizing and gasefying the oil during the suction and compression strokes in which the compression pressure attained

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is less to any desired degree than that required in the purely Diesel engine, though not less than is common in the ordinary well-known types of gasoline (petrol) engines, which employ mechanically-timed and controlled electric ignition.

Even the most modern forms of Diesel engines, whether of the two or the four-cycle type, are constitutionally large, heavy, complicated, and very costly in comparison with a non-Diesel oil engine. They require a great many expensive auxiliaries and accessories new to the marine engine trade, such as high-pressure compound intercooled air compressors from which air at about 1,000 pounds pressure is pumped into heavy drawn steel tanks or bottles for use in fuel injection or atomization and for starting the engine. There are also required governor-controlled variable-stroke high-pressure fuel pumps, usually one to each working cylinder, fuel injection timing mechanisms for the delivery of the high-pressure oil and air to the workinng cylinders, delicate and easily-clogged fuel injecting and atomizing valves, one in each working cylinder, involving considerable complex high-pressure piping, and several pressure gages. In addition to the difficulties of design and high cost of construction and maintenance involved in the purely Diesel type engine, the eternal vigilance required by the operating engineers to insure continuous and reliable operation of so much high-pressure equipment can be fully appreciated only by those who have had such problems to solve and deal with.

In some of the multi-cylinder two-cycle Diesel marine engines, there is also the further complication of a large double-acting scavenging air pump, usually located on the forward end of the engine. This large low-pressure air compressor involves an extra crank and, therefore, an appreciably longer and more expensive engine crank shaft, a crosshead with guides, a piston rod and stuffing box, two multiplex nests of large area, intake and discharge disk valves, and a low-pressure air receiver. In other two-cycle Diesel engines, the pistons in each of the working cylinders are of the two-diameter or differential type, the lower or large end acting as a low-pressure scavenging pump for its respective working cylinder, thus involving a higher and heavier engine, bulky cylinder castings and a double diameter of cylinder bore and multiplex nests of intake and discharge valves.

Besides the weight, bulk and cost of such complications there is always in the Diesel engine the liability of at least temporary inability to operate, due to defects, wear and small changes of adjustment of any of the detail parts of such mechanisms. Although the Diesel engine has no need of a carbureter, the multiple pumps, injectors and atomizers, high-pressure piping, air bottles, gages, etc., are a vastly more complicated and costly substitute. In the past Diesel engine builders have boasted that their engines needed no electric or other forms of ignition appliances, but now some of the more advanced investigators are recommending the incorporation of electric ignition apparatus in the designs in order to synchronize and time more accurately the ignition point in all cylinders to eliminate the dangers due to destructive premature ignitions and to insure positive ignition in case of excessive leakage and loss of compression resulting from wear, breakage or gumming up of the valves, pistons or piston rings.

In spite of the fact that the fuel consumption per brake horsepower hour in Diesel engines is remarkably low (when everything is operating properly) and that the available fuels are remarkably cheap and plentiful, there are other considerations involved which may more than offset these apparent economies. The principal consideration is the high first cost of the engine, and the cost of repair and upkeep. As compared with a non-Diesel oil engine installation, there is an increased expense of a larger operating crew, together with their necessary quarters, and above all there is the heavy overhead cost due to possible necessity of laying up the ship for cleaning, repairs and readjustment of any derangement of the machinery, which might come about through the shortcomings of the operating crew, and their lack of sufficient understanding of the principles of operation of such engines and their accessories. Moreover, the ability of the Diesel engine to burn any and all kinds of fuel oil continuously and reliably is for the present at least much overrated. The writer has seen chunks of carbon, practically coke, larger than a man's fist, taken out of the combustion spaces of Diesel engines operating on paraffin base oils of lighter weight and density than 28 degrees Baumé. This occurred simply because the problem of perfect oil atomization in the engine cylinder is if anything even greater in the Diesel engine than in the lower-compression non-Diesel type. With all the infinite care evidenced in the design and construction of the Diesel fuel injectors and atomizers, the oil is seldom completely atomized. The first and last drops of each fuel charge are especially liable to enter the engine cylinder in a liquefied form, and any engine, whether of the Diesel or other type, which permits oil fuel to enter the cylinder in liquid form will "crack" such oil, gasifying the lighter hydrocarbons which it contains and baking the remainder into hard gritty carbon which accumulates until the engine becomes inoperative. It is absolutely imperative for efficiency, continuous and reliable operation that all oil to the last atom be so finely subdivided or atomized upon its admission to the engine cylinder that even without the application of the heat of compression the cloud of vapor thus produced will remain suspended in the air charge for several minutes.

Instead of its many significant faults and drawbacks, the Diesel engine as a prototype of what the future will undoubtedly produce, has given abundant evidence of the latent possibilities of the internal combustion engine, and has fully warranted further experiment. The excessive cost of experimenting with that form of engine, however, which must be paid by each and every concern that attempts such construction, compared with that found necessary in the development of gas and gasoline (petrol) engines, has very naturally scared off the very conservative American engine builders from research along Diesel engine lines. Although anxious to keep step with Europe in the matter of the number and size of deep sea oil engine installations actually made, nevertheless, to interest American manufacturers at all, the forthcoming American product will have to involve less cost of experiment and research, less weight, size, bulk and complication, require a less scholarly prepartion on the part of engineers, be more easily and cheaply designed and constructed, and, in short, constitute itself more of a commercially practical proposition than has been the case thus far with the orthodox Diesel engine.

In the search for such a production, most American builders have in turn experimented with, developed and finally rejected several distinct methods of oil combustion with almost infinite variations and modifications of detail. Practically all of these experiments have aimed to utilize with as little change as possible the builders' stock models of gas and gasoline (petrol) engines. Some have attempted to pass the oil fuel through the more or less regular gasoline (petrol) carburetor, either hot water jacketing the outside of the bowl, the passage surrounding the nozzle, the inlet pipe from the carburetor to the engine inlet valves, or all of these. Also separately or in conjunction with this water jacketing, they have taken the incoming air from a heater sleeve, surrounding the exhaust pipe, and in some cases have placed an exhaust heated radiator or nest of tubes within some portion of the inlet pipe. Very few of these, however, made any serious attempt at a nozzle which would efficiently blow the fuel into veritable atoms, though many incorporated various screens, fans, helices and other devices over the nozzles with the intention of breaking up the spurt of fuel. Practically all

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of these, however, serve only to set up excessive fluid friction and a high vacuum in the intake pipes, causing unnecessary loss of engine efficiency and power output.

Although such carburetors when equipped with properly-regulated automatic air valves produce very good results when operating on heavy gasoline (petrol) and the lighter kerosenes (paraffines), nevertheless as the oils used through them become heavier their degree of successful operation rapidly drops. With fuels no heavier than "Solar" oil and "gas" oil they become commercially impossible. The engines lose all flexibility of speed and power control, the exhaust proves extremely sooty, the pistons, piston rings, valves and igniters become rapidly tarred and gummed up, the piston spaces are thickly carbonized, self and pre-ignition becomes so troublesome that the compression pressure must be reduced to a most uneconomical point and the consumption of fuel per brake horsepower hour is about double of what the consumption of gasoline (petrol) would be in the same engine. The power output is considerably less, and finally after from ten to fifteen hours' operation the engine becomes inoperative, requiring a thorough cleaning.

The carburetor method, though commercially practical with gasoline (petrol) and the lighter kerosenes (paraffines), can offer very little hope for the future on heavier oil fuels; for, though it may be possible to so design the nozzles as to atomize sufficiently the fuel charges passing through them, the succeeding length of inlet pipe to the engine valves with its several necessary bends and elbows will assuredly condense or reliquefy on the walls of such passages sufficient oil fuel to make the fouling of the combustion chambers apparently certain. However extensive the hot water jacketting and however carefully the exhaust heated portions of such a carburetor may have been designed and placed, it is practically impossible to so control automatically the applied heating effect of such appliances under all the wide variations of load and speed as to make it always just sufficient to gasify even a properly-atomized oil spray. Either this heat will at times prove sufficient to "crack" the oil into its lighter and its heavier components, the heavy portions being thrown down on the walls of both the inlet passages and the combustion chambers, or it will prove so low as to permit reliquefaction on these walls. Both of these conditions cause fouling, non-reliable operation, high fuel consumption, and a low power output.

As the main idea of oil engine research is to make available in the engine cylinder the cheaper and more abundant distillations from crude petroleum in the place of the very satisfactory though very limited available supply of light and heavy gasoline (petrol) and the lighter kerosenes (paraffines), the so-called carburetor type of oil engine, because of its restriction to the lighter distillates only, holds little promise for the future. Such lighter hydrocarbon distillates from crude petroleum constitute not over 10 percent of volume of the crude oil as taken from the ground. The market cost of the gasoline (petrol) and kerosene (paraffin) distillations, in the absence of a material market for the other 90 percent, must of course cover the entire cost of production, refining, handling and marketing of all the raw product, minus what may be received for the heavier by-products, which at present is comparatively little. Should the production of a successful type of heavy oil engine be accomplished, the market value of the lighter petroleum distillations would drop very considably and the price of the heavier distillations would rise slightly, tending to equalize the price on all by-products and making an enormous increase of internal combustion engine fuels available at a reasonable price. This no doubt would result in a widespread growth of all branches of the internal combustion engine industry and make possible marine engines for the largest of the sea-going ships.

(To be continued.)

Further Experiments Upon Wake and Thrust Deduction*

BY W. J. LUKE

The following notes on wake and thrust deduction are supplementary to the paper on the same subject presented to this Institution in 1910. At that time the modifications in the hull-efficiency elements of a model, 204 inches long, 30 inches wide, 9 inches in draft, and of 0.65 block coefficient, due to certain variations of linear speed, propeller diameter, pitch ratio, area ratio, etc., formed the subject matter. Briefly stated, the results then presented show that increase in speed caused a slight decrease in wake and hull efficiency, but that the variation was of little moment. A considerable decrease in wake and increase in thrust deduction followed increase in diameter in single-screw experiments. The hull efficiencies with the larger screws were thus substantially less than those with the smaller.

In twin-screw experiments, with the centers of shafts in a standard position, the larger the screws were the greater became the wake and hull efficiencies when outward turning. When inward turning, the wake values were modified only slightly with increase in diameter, but a steady increase in thrust deduction gives a much lower hull efficiency value with the larger diameter screws. It was shown by the experiments with various transverse positions of the screws that the closer these were placed to the hull the higher became the wake and hull efficiency values. Experiments dealing with fore-and-aft position indicated that the further aft the screws were placed the less became the wake and thrust deduction values. Decrease in the value of the hull efficiency elements accompanied decrease in pitch ratio, and neither area nor number of blades had any appreciable effect on wake and thrust deduction.

The most important point disclosed by the results then presented was the great effect of direction of rotation when the angles of shaft bossings were extreme—*i. e.*, when horizontal or vertical. With the standard screws a wake fraction of only 0.10 was obtained when revolving inwards behind horizontal bossings, while this was increased to 0.32 when outward turning, the hull efficiency value obtained in the former case being 0.94, while 1.10 was recorded in the latter. The resistance of the bossing itself increased with the angle of inclination either way from the normal—that is, it was least when at an angle normal to the line of the shell, and greater when horizontal or vertical. The additional hull efficiency gained with the proper direction of rotation, when extreme angles such as the horizontal or vertical were adopted, almost compensated for the greater resistance of the bossing. Apart, however, from this consideration, the results indicated the possibility of committing a gross error by the adoption of inward-turning screws with horizontal bossings, in which bossings causing great resistance would be associated with low hull-efficiency values.

During the discussion that followed the reading of the paper referred to, Professor Welch asked whether the investigations had been extended to cases of screws having a common axis, one screw being placed behind the other, such as the contrary-turning screws of locomotive torpedoes. At that time apparatus for experiments in this direction had been prepared, but no substantial series of results had then been obtained. The purpose of the present communication is to give the results of experiments with contrary-turning screws on a common axis; of experiments with tandem screws, and also of experiments with quadruple screws.

Lieutenant-Colonel Rota, in his paper, "The Propulsion of Ships by Means of Contrary-Turning Screws on a Common

* From a paper read before the Institution of Naval Architects, London, April, 1914.

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Axis," delivered to this Institution in 1909, drew attention to the success attending the adoption of that system in a 46-foot steamboat of 25 tons displacement. The results communicated showed an advantage in the adoption of contrary-turning screws of about 25 percent over the performance of a single screw of larger diameter. The author suggested this to be due in the main to two causes, viz., increase in hull efficiency through increased wake and decreased thrust deduction, and increase in propeller efficiency. With this information and previous experimental work in view, the present experiments were directed towards ascertaining what efficiency would be exhibited by contrary-turning screws on a common axis (hereafter designated "combination" screws) apart from the introduction of the hull-efficiency elements; and further, what influence was exerted by these latter factors. It was considered that "open" and "behind" experiments, such as are usual when undertaking wake investigations, would give the necessary information to enable a judgment to be formed as to wherein lay the virtue of the "combination" screws. Records of thrusts, torques, revolutions, and speed of advance were obtained with various combinations of screws, when working in the "open" and also behind models. Two models were brought into requisition, one being that on which the investigations already brought before your notice were made, the other being of almost similar dimensions, but of finer block coefficient.

PARTICULARS OF MODELS.
(Dimensions in Inches.)

	Length.	Breadth.	Draft.	Block Coefficient.	Displacement in Pounds in Fresh Water.
Full model....	204	30	9	0.65	1296
Fine model....	200	30	9	0.60	1175

The standard propeller chosen was that adopted in the former experiments—namely, of 6 inches diameter—having three blades, 1.2 pitch ratio, and 0.375 disk area ratio.

Experiments were made first with a single standard screw, and its performance was noted when revolving "behind" the full model when advancing at a speed of 332 feet per minute (corresponding to a speed of 16 knots for a 400-foot ship), and in the open at a speed of 280 feet per minute (estimated to be a suitable speed allowing for wake). With the combination system experiments were made with the standard screw in conjunction with each of three others identical with the standard in all particulars except pitch ratio, of which the three values used were 0.8, 1.2, and 1.6, the standard screw leading in each case. Were there any inherent virtue in the "combination" screws apart from that accruing from their interaction with the model, it would be apparent in efficiency when running in the open. It was found that the two propellers of 1.2 pitch ratio in combination gave a thrust which was approximately twice that obtained by the single screw; the torque, however, was perceptibly greater than twice that required to drive the single screw.

With an ordinary twin-screw arrangement the thrust obtained equals twice that of one screw, and the propeller efficiencies are identical. The open efficiency of the "combination" screws at the point of self-propulsion was found to be about 0.86 of those of the ordinary twin-screw. The results obtained from the experiments behind the full model (when investigating wake and thrust deduction values) were so remarkable that the experiments with the two standard "combination" screws were repeated three times. Wake experiments had previously been made with the ordinary twin-screw arrangement, and also with the single screw (as described in the previous communication), and the results obtained were as follows:

	Wake.	Thrust Deductions.	Hull Efficiency.
Twin screws	0.20	0.15	1.02
Single screws	0.34	0.17	1.11

With the "combination" screws the wake was discovered to be greatly in excess of that obtained by the single screw, while the thrust deduction was very little different; accordingly the hull efficiency showed a considerable increase. The results of the four separate sets of experiments with the two standard screws are as follows:

	Wake.	Thrust Deductions.	Hull Efficiency.
1.	0.57	0.17	**1.31**
2.	0.61	0.17	**1.34**
3.	0.61	0.17	**1.34**
4.	0.64	0.19	**1.33**

Experiments with the 1.2 and 0.8 pitch ratio screws in combination, and with the 1.2 and 1.6 pitch ratio, confirmed the results obtained with the standard screws. Together with the average value of the four cases above, they stand thus:

Pitch Ratios.	Wake.	Thrust Deduction.	Hull Efficiency.
1.2 and 0.8	0.59	0.19	1.27
1.2 " 1.2	0.61	0.17	1.34
1.2 " 1.6	0.57	0.19	1.27

The efficiencies of the screw when working both in the "open" and "behind" were observed, and no appreciable difference was evident in the rotative efficiencies. Because of the great increase in the hull-efficiency values, the "combination" screws show a considerable gain in propulsive efficiency over the ordinary twin-screw arrangement. In the latter case, the propeller efficiency at the point of self-propulsion was 0.56, and the hull efficiency 1.02. Thus propeller efficiency $\times$ hull efficiency $= 0.57$. With the "combination" arrangement the propeller efficiency at the point of self-propulsion was 0.49, and the hull efficiency 1.34, giving $0.49 \times 1.34 = 0.66$, as compared with 0.57 in the ordinary twin arrangement. The above results are for the full model.

Four years elapsed between the experiments with the full and fine models. The relative efficiencies of the single screw and the "combination" screws were again put to test, and the former experiments were verified, in that the open experiments showed the efficiency of the "combination" arrangement to be 0.86 that of the single screw, and accordingly also that of the ordinary twin screws. With the finer model, when advancing at a speed of 372 feet per minute (corresponding to a speed of 18 knots for a 400-foot ship), the same peculiarities as to high wakes and hull efficiencies with the "combination" screws, as with the fuller form, are evident, but they are not so much accentuated. With the single screw and the ordinary twin arrangements, the results obtained were as follows:

	Wake.	Thrust Deductions.	Hull Efficiency.
Single	0.22	0.16	**1.02**
Twin	0.13	0.13	**0.98**

The same combinations of contrary-turning screws were run as with the full model, with the following results:

Pitch Ratios.	Wake.	Thrust Deduction.	Hull Efficiency.
1.2 and 0.8	0.29	0.12	1.14
1.2 " 1.2	0.33	0.12	1.17
1.2 " 1.6	0.33	0.12	1.17

The gain in this case is not nearly so great as with the full model.

The open experiments show that, apart from wake effect, the system is not as good as a twin-screw arrangement. With a full-ended model, as used in the first experiments, the wake is such that the combination screws are able to make sufficient use of it to nullify the loss in open efficiency, and to show a net gain of 15 percent. In this fine model, however, the wake is diminished so that the gain possible in hull efficiency just about balances the open efficiency loss, and the gross return, propeller efficiency $\times$ hull efficiency, practically equals that of the twin-screw arrangement.

The results obtained by the "combination" screws when behind were so remarkable that it was thought that tandem

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propellers might also show peculiar propensities. Previous experiments undertaken at Clydebank had shown what had also been discovered by Professor Durand, that in the open tandem screws did not give a thrust equal to that of twin screws of the same dimensions. On placing upon a single shaft two similar screws placed two-thirds of a diameter apart or more under the same circumstances of revolution and slip, the thrust was found to be only about 10 percent greater than when one of the two screws was removed; the efficiency of the system was also found to be low.

Experiments with tandem screws in the open and when behind the finer model were carried out; the screws used in these experiments were in case (*a*) two of 1.2 P.R., and in case (*b*) of 1.2 P.R. and 1.6 P.R. The 1.2 P.R. screw was leading in both cases, and the propellers were placed quite close to each other, just as they were in the combination experiments. From the results obtained the former conclusion that tandem propellers have little or nothing to recommend them was amply confirmed. In the open the thrusts obtained are low compared either with a twin or combination arrangement. The open efficiency of the latter is not great, but the hull efficiency elements associated with "combination" screws are high. The tandem screws have no such effect on wake and thrust deduction, and the hull efficiency values obtained were not abnormal, as the following figures show:

Pitch Ratios.	Wake.	Thrust Deduction.	Hull Efficiency.
1.2 and 1.2	0.26	0.20	1.01
1.2 " 1.6	0.24	0.16	1.04

The hull efficiency values are thus about 15 percent less than those obtained with the contrary-turning screws, with the same combination of pitches.

Taking the hull efficiencies in conjunction with the propeller efficiencies at the point of self-propulsion, the following figures are obtained for the fine model:

TANDEM SCREWS.

Pitch Ratios.	Hull Efficiency.	Propeller Efficiency (at point of self-propulsion).	Hull Efficiency × Propeller Efficiency.
1.2 and 1.2	1.01	0.50	0.50
1.2 " 1.6	1.04	0.44	0.46

These compare with the undernoted corresponding figures for the "combination" arrangement:

Pitch Ratios.	Hull Efficiency.	Propeller Efficiency (at point of self-propulsion).	Hull Efficiency × Propeller Efficiency.
1.2 and 1.2	1.17	0.58	0.68
1.2 " 1.6	1.17	0.52	0.61

It will be observed that the propeller efficiencies of the "combination" screws at the point of self-propulsion are (in this case) considerably greater than those of the tandem arrangement. This is accounted for by the fact that to reach the point of self-propulsion the propellers were working at a high real slip, and that the combination screws gave the required thrust with fewer revolutions than did the tandems, and at higher points in the efficiency curves. If the tables giving the comparative open efficiencies of the combination and tandem screws be examined (given in the Appendix) it will be seen that, although at moderate slips the tandem screws are the more efficient, at the higher slips this superiority disappears. At the real slip requisite to overcome the resistance of this particular model, the efficiencies of all the combinations of the contrary-turning screws are considerably in excess of those of the tandems.

To sum up this question, it appears that the "combination" screws have not a little to recommend them, and, were the engineering difficulties connected with their application to marine propulsion overcome, they would be well worthy of consideration. Tandem screws have nothing to recommend them.

With the introduction of triple and quadruple screws for the propulsion of ships arose the necessity for modifying the existing experimental apparatus so as to be able to undertake wake and thrust deduction experiments on models of vessels with more than two shafts. Before these modifications were introduced the experiments with the forward and after screws were made separately just as if each set were a pair of twin screws; the condition taken for analysis of each pair being that the thrust equals half the resistance of the model with no screws working behind plus the augment caused by these screws. The aggregate thrusts of the forward and after screws thus equal the total augmented resistance. This assumes complete non-interference. This latter was a doubtful point and it was resolved to undertake experiments in the first place, as for twin screws, and afterwards to observe the effect on the performance of the forward screws of a pair of revolving but non-recording screws placed in the after position, and also on the after screws of a pair placed in the forward position; these non-recording screws were placed on a fixed frame, and could be given any desired rotary speed.

The results given apply only to the fine model. Three-bladed propellers, 4 inches in diameter, of 1 pitch ratio and of 0.6 disk area ratio, were used in these experiments. The inner propellers were placed just forward of the after perpendicular, with the tips reasonably clear of the hull; the outer propellers were placed so that in projection their disks just touched the disks of the inner screws. The outer screws were carried as far forward as the form of the ship would allow, giving the same clearance from the hull as obtained in the after pair.

The results of the twin-screw experiments call for no special comment except that the high hull efficiencies obtained are due probably to the close proximity of relatively small propellers to the hull of a very wide model.

From these results the action of the forward screws has a material influence on the wake of those in the after position, and the hull efficiency of these screws is considerably reduced. The suction of the after screws reduces very slightly the wake proper to the forward screws, and the hull efficiency elements remain practically the same as in the twin-screw experiments. Broadly speaking, the wakes of the after screws were reduced on the average by 0.07, and those of the forward screws by 0.01. Some light upon two most important points in connection with the propulsion of vessels by quadruple screws is obtained from these experiments. In the first place, the wake value to use in propeller design is approximated to, and, in the second place, the combination of directions of rotation to ensure the highest gross hull efficiency can be obtained. The various combinations possible are:

(*a*) After screws, outward turning; forward screws, outward turning.

(*b*) After screws, outward turning; forward screws, inward turning.

(*c*) After screws, inward turning; forward screws, outward turning.

(*d*) After screws, inward turning; forward screws, inward turning.

With this particular model the gross hull efficiency values obtained with these various combinations are almost identical, although it is not possible to say that this is a general rule. The experimental conclusions may at least be some guide in the analysis of trial results, which analytical conclusions will form the basis for design in future cases.

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The Design of Merchant Ship Forms*

BY PETER DOIG

Load Waterline Shape

For the determination of load waterline half-breadths Table V has been prepared. The values were found by dividing the waterline breadths (in terms of the extreme breadth) by the area coefficients at the respective stations as given in Tables II, III and IV. These ratios give contours of waterline, ensuring good forms for propulsion while conforming to the other important elements in a good design. The procedure in designing the lines of a ship is somewhat as follows:

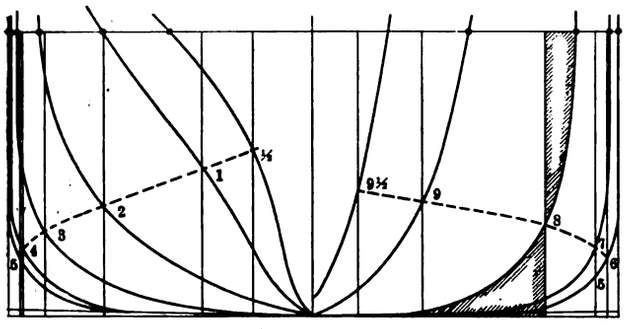

Fig. 2

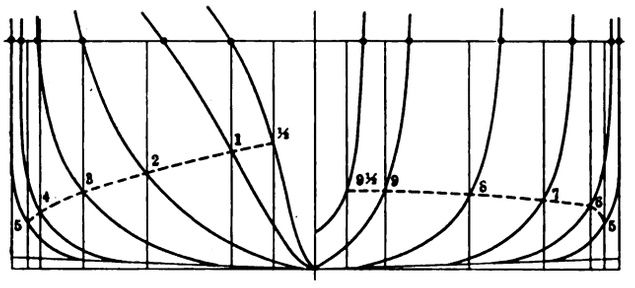

Fig. 3

When (1) length, (2) breadth, (3) draft, (4) displacement, and (5) longitudinal position of center of buoyancy are known, a curve of transverse sectional areas is constructed to these requirements. The beam, draft and rise of floor lines are drawn on a suitable scale as the foundation of the proposed body plan. To obtain sections of the required areas, rectangles bounded by the waterline, center line, base and a line from the waterline to the base are laid off (Fig. 2). The outlines of the transverse sections are drawn so that the parts inside and outside the bounding line, as marked in the figure, are equal, thus producing the areas required.

These are in substance the necessary steps in designing forms by the use of the tables appended. Fig. 2 represents the lines of a freight and passenger steamer of the subjoined particulars:

Length, 500 feet.
Breadth, 63 feet.
Draft, 29 feet 6 inches.
Displacement, 18,400 tons.
L. C. B., 1.5 feet forward or 0.3 percent of length.

This gives a block coefficient of .692, which with a mid-area coefficient of .960 corresponds to a prismatic coefficient of .721. From Tables II and III the area coefficients are obtained by linear interpolation, as in the first column below. The figures in the second column, from Table V, are multipliers to be applied to these area coefficients, giving the waterline coefficients, or breadth at station ÷ extreme breadth, in column 3. The last column gives, by multiplication of the values in the first by the mid-area coefficient ($C_m = .960$) the mean breadths of the sections in terms of the ship's half breadth amidships:

Station	1	2	3	4
0	0	...	0	0
½	.199	2.32	.462	.191
1	.380	1.82	.692	.365
2	.717	1.26	.904	.688
3	.919	1.05	.965	.882
4	.998	1.00	.998	.958
5	1.000	1.00	1.000	.960
6	1.000	1.00	1.000	.960
7	.964	1.01	.974	.926
8	.793	1.09	.866	.762
9	.374	1.36	.509	.359
9½	.150	1.61	.242	.144
10	0	...	0	0

The widths of the load waterline and of the rectangles of equivalent sectional area are figured from the last two lines of above table and laid off to scale (Fig. 2).

To obtain an approximately "fair" body plan the locus of the crossing points of the sectional outlines and the sides of the rectangles is kept in as gradual and sweet a curve as possible. When this has been done the resulting plan will be ready for final fairing in the usual drawing-office manner. Fig 3 represents the form of a transatlantic liner 750 feet by 85 feet by 32 feet draft, of 34,300 tons displacement, and a position of center of buoyancy 6.0 feet aft of amidships.

The body plan of Fig. 4 is that of a slow freighter 420 feet by 56 feet by 28 feet 6 inches draft, 15,450 tons displacement and a center of buoyancy 2.0 feet forward of the center between perpendiculars.

Metacentric Height Tables

In the preliminary stages of a design it is necessary to fix quickly the beam requisite for the desired "G M." The height

Table V.

Prismatic Coefficient.	Station. ½	1	2	3	4	5	6	7	8	9	9½
.52	2.19	2.09	1.78	1.36	1.11	1.00	1.06	1.20	1.37	1.56	1.68
.54	2.09	2.01	1.68	1.31	1.09	1.00	1.05	1.17	1.32	1.49	1.60
.56	2.01	1.93	1.59	1.25	1.07	1.00	1.04	1.14	1.26	1.42	1.54
.58	1.95	1.84	1.49	1.20	1.05	1.00	1.03	1.11	1.21	1.36	1.47
.60	1.91	1.77	1.41	1.16	1.03	1.00	1.02	1.09	1.16	1.30	1.42
.62	1.90	1.71	1.33	1.13	1.02	1.00	1.02	1.07	1.13	1.25	1.40
.64	1.93	1.70	1.30	1.11	1.02	1.00	1.01	1.05	1.12	1.24	1.41
.66	2.03	1.74	1.28	1.09	1.01	1.00	1.01	1.04	1.11	1.26	1.44
.68	2.19	1.82	1.28	1.08	1.01	1.00	1.01	1.03	1.11	1.31	1.51
.70	2.32	1.85	1.27	1.07	1.00	1.00	1.00	1.02	1.10	1.36	1.59
.72	2.32	1.82	1.26	1.05	1.00	1.00	1.00	1.01	1.09	1.36	1.61
.74	2.30	1.74	1.21	1.03	1.00	1.00	1.00	1.01	1.07	1.34	1.58
.76	2.20	1.58	1.15	1.02	1.00	1.00	1.00	1.00	1.06	1.30	1.52
.78	2.11	1.47	1.09	1.01	1.00	1.00	1.00	1.00	1.05	1.25	1.46
.80	2.04	1.41	1.05	1.00	1.00	1.00	1.00	1.00	1.03	1.22	1.41
.82	1.98	1.36	1.03	1.00	1.00	1.00	1.00	1.00	1.02	1.19	1.37
.84	1.93	1.31	1.01	1.00	1.00	1.00	1.00	1.00	1.01	1.18	1.34

Table VI.

Cp	r	LCB 1% Forward.	LCB Amidships.	LCB 1% Aft.
.52	.823	.071	.075	.0785
.54	.841	.0695	.0725	.0750
.56	.860	.0690	.0705	.0720
.58	.870	.0680	.0690	.0700
.60	.880	.0685	.0680	.0680
.62	.890	.0685	.0680	.0680
.64	.890	.0695	.0680	.0685
.66	.890	.0705	.0700	.0715
.68	.890	.0730	.0725	.0745
.70	.890	.0750	.0745	.0770
.72	.890	.0770	.0765	.0790
.74	.895	.0785	.0775	.0800
.76	.903	.0795	.0785	.0800
.78	.914	.0805	.0795	.0800
.80	.925	.0815	.0805	.0800
.82	.937	.0820	.0805	.0795
.84	.948	.0825	.0810	.0790

* Concluded from the April issue.

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of the center of gravity of a proposed ship can be closely gaged by proportioning to a similar vessel by depths of hull. To estimate vertical distance of metacenter above base, the "*B M*" or distance between center of buoyancy and metacenter and the height of the center of buoyancy above base, must be known. In obtaining "*B M*"

$$\text{which} = \frac{\text{Inertia of waterplane}}{\text{Volume of displacement}}$$

the inertia of waterplane can obviously be closely estimated for a design to be drawn by the use of the foregoing tables. As volume of displacement for a given value of C_p is proportional to the value of C_m adopted, we get the general expression:

$$B\,M = \frac{I}{c\,C_p\,C_m},\ c \text{ being a constant dependent on dimensions.}$$

Since $B\,M = \dfrac{a\,L\,B^3}{L \times B \times d \times C_b} = \dfrac{a\,L\,B^3}{L \times B \times d \times C_p \times C_m} = a\,\dfrac{B^2}{d\,C_p\,C_m}$

where a is a coefficient, L, B and d being length, breadth and

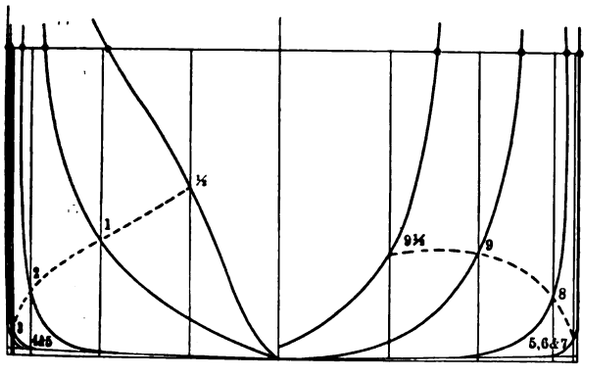

Fig. 4

draft, respectively, we get the formula $B\,M = K\,\dfrac{B^2}{d\,C_m}$ for any particular value of C_p.

The vertical position of center of buoyancy can be very closely figured by Morrish's formula, which may be expressed,

$$V.\,C.\,B. = \left(.833 - .333\,\frac{C_p\,C_m}{C_w}\right) d.$$

C_w being coefficient of waterplane or area of waterplane ÷ length times breadth. Substituting r for $\dfrac{C_p}{C_w}$, the formula becomes $V.\,C.\,B. = (.833 - .333\,r\,C_m)$.

In Table VI the values K and r in these formulæ are given for varying values of C_p and position of center of buoyancy longitudinally, which latter affects waterplane inertia to some extent, but only very slightly alters $V.\,C.\,B.$

The use of the values K and r may be best illustrated by the following examples. A liner 650 feet by 73 feet by 49 feet to awning deck, 30-foot draft and 29,000 tons displacement, $L.\,C.\,B.$ being 0.5 percent of length aft, has to have $C\,M$ of 2.0 feet. The lines are designed by the foregoing methods, using a C_m of .960. From a similar vessel, whose $C.\,G.$ is known by inclining experiment, the height of center of gravity is estimated as 29.5 feet above keel. This entails a height of metacenter above base of 31.5 feet. Testing the proposed beam of 73 feet, we get $C_b = .713$; $C_p = \dfrac{.713}{.960} = .744$.

From Table VI, $r = .897$, $K = .079$.

Hence $V.\,C.\,B. = (.833 - .333\,r\,C_m)\,d.$
$= [.833 - (.333 \times .897 \times .960)]\,d.$
$= (.833 - .286)\,d = .547\,d = .547 \times 30.$
$= 16.4$ feet.

$$B\,M = .079\,\frac{B^2}{d\,C_m} = .079 \times \frac{73^2}{30 \times .960} = 14.65 \text{ feet.}$$

M above base $= 16.4 + 14.65 = 31.05$ feet.

It is thus evident that either depth will have to be decreased, bringing down $C.\,G.$, or beam increased, which will give greater "M above base." Adding 2 feet to beam, which is the more practicable alternative, the calculation results (keeping displacement constant):

$C_b = .694$; $C_p = .723$.
$V.\,C.\,B. = 16.45$ feet above base.
$B\,M = 15.0$ feet.

31.45 M above base.

$G\,M = 31.45 - 29.5 = 1.95$ feet.

For design purposes this approximation is sufficiently accurate and rapid.

Electric Towing Locomotives for Panama Canal Locks

The first lot of electric towing locomotives for hauling vessels through the locks of the Panama Canal are now being received at the Isthmus. In all forty of these "electric mules" are being built by the General Electric Company, Schenectady, N. Y., for this purpose. The machines weigh 82,500 pounds; measure 32 feet 2½ inches long by 8 feet wide by 9 feet 3 inches, the greatest height over the cabs; have an available tractive effort as high as 47,500 pounds, and a windlass rope pull of 25,000 pounds. Four of these locomotives, two on each side, will ordinarily tow steamships through the locks. Sometimes six engines will be needed to handle extra large vessels; in every case two astern, acting as a brake on the ship's movements, will give direction to her course. No vessel will be allowed to enter the locks and go through under her own power.

The locomotive is built up of cast steel side and end frames, cross ties and bedplates. It is mounted on two axles with wheels in accordance with MOB standards. The entire frame is supported from journal boxes of the regular railway type by means of coiled springs. The sides and top of the body are enclosed by sheet iron covers which fit in place and are very easily removable. At each end are enclosed cabs so that the locomotive may be operated from either end.

The locomotive is propelled by means of a rack rail while towing, and while going up or down the steep grades from one level to another at a speed of 2 miles per hour. While running idle or on return tracks, the speed is changed to 5 miles per hour, and the machine is propelled by the regular traction method, the rack pinion being entirely released This change is effected by manually-operated clutches located in the gear mechanism in connection with a lever in each cab similar to a steam locomotive.

The locomotive is driven by two 75-horsepower totally-enclosed motors of the mill type, one being direct-connected through reduction gearing to each axle. Three-phase, 25-cycle, 220-volt current is used and is collected by contact plows.

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The motor and traction gearing is mounted on a common baseplate, which in turn is mounted on a driving axle and spring suspended to the locomotive frame the same as in regular railway practice.

In the center of the locomotive is located a vertical windlass with drum, the capacity of which is 800 feet of 1-inch steel hawser cable. The windlass, with its driving motors and gearing, is mounted on a solid baseplate, and is likewise independent of the movement of the locomotive frame. The cable drum extends above the locomotive cover and has a floating guard placed around it to retain the cable while coiling loose.

The windlass cable is handled by two 20-horsepower motors, also totally enclosed and of the mill type. One is geared for a rope speed of 12 feet per minute at a pull up to 25,000 pounds at 2-foot radius, and its function is to adjust the position of the ship for anchor or while being towed through the locks. The other motor is geared for a rope speed of 200 feet per minute at 2-feet radius, and its duty is to take up slack or pay out cable or wind in any part of the entire length of cable as may be required. The cable drum is driven thorugh a friction device, which can be set at any desired value from zero up to the full capacity of the motor.

The traction motors as well as the windlass motors are controlled from either cab. In other words, the control equipment is duplicated. The two traction motors are operated by one master controller and contactors forward and reverse, while the windlass motors are operated by a reversible drum controller, and the clutch on the main vertical shaft by a solenoid.

Oil Tank Steamer Frank H. Buck

The First Longitudinally Framed Vessel Built on the Pacific Coast—Description of the Hull and Machinery

The *Frank H. Buck,* a steel single screw tank steamer, built for the Associated Oil Company by the Union Iron Works, San Francisco, Cal., is the first tank steamer built on the Pacific Coast on the Isherwood system. The keel of this vessel was laid on August 21, 1913. She was launched Feb. 11, 1914, and will be completed about April 15.

The ship has been constructed in accordance with Lloyd's requirements to Class 100 A 1 in Register. The principal dimensions and particulars of the vessel are as follows:

Length over all	426 feet 9 inches
Length between perpendiculars	410 feet 0 inch
Beam, molded	55 feet 3¾ inches
Depth, molded to upper deck	31 feet 8 inches
Load draft	27 feet 0 inch
Load displacement	13,960 tons
Cargo capacity	63,964 barrels
Fuel capacity	2,211 barrels
Gross tonnage	about 5,900
Engines	3-cylinder triple-expansion
Diameter of cylinders	26½, 45, 75 inches
Stroke	48 inches
Condenser cooling surface	3,755 square feet
Revolutions per minute	65
Designed indicated horsepower	2,600
Designed speed	10½ knots
Boilers	Four Scotch, single end
Length of boilers	11 feet 9 inches
Diameter boilers	14 feet 0 inch
Working pressure	200 pounds
Number of furnaces each boiler	3
Propeller	One four-bladed
Propeller, diameter	18 feet 9 inches
Propeller, pitch	18 feet 9 inches
Propeller, helical area	99.41 square feet

The hold is subdivided into tanks for carrying oil in bulk, the starboard and port compartments being separated by an oil-tight centerline bulkhead up to the top of the expansion trunk. The 'tween decks in the wings outside the expansion trunk is arranged for carrying ordinary freight. Fresh water is caried in a double bottom under the engines and boilers. A double bottom is also fitted forward for carrying either fresh water or ballast. The fore peak and after peak are also constructed for carrying fresh water.

The vessel is constructed with straight stem and elliptical stern, with two continuous decks and a raised forecastle. An open bridge is built amidships and a full poop aft. The vessel is rigged with three steel pole masts, with three cargo booms on the fore mast, two on the main mast, and one on the mizzen mast. Accommodations in the poop are fitted up for the engineers, stewards, firemen, seamen, etc. Upon the bridge deck amidships, enclosed in a steel house, accommodations are fitted up for the captain, deck officers and steward, consisting of a pantry, dining room, staterooms, bath room and toilets. Above this house and on a level with the flying bridge a teak wheel house and chart house are built. The vessel is provided with a No. 5 Shaw Spiegle towing machine installed aft, a Brown steam tiller, steam windlass and capstan, two powerful warping winches and two warping capstans.

Hull Construction

The vessel is constructed on the Isherwood system, with transverse framing abaft the engine room. The keel is of the flat plate type furnaced out at each end to properly meet the heels of the stem and stern frames. The stern is of wrought steel and the stern frame of cast steel in two sections, with well-proportioned scarphs. The rudder is of the single-plate type, the frame of forged steel with arms keyed to the stock, while the stock is in two pieces, with a coupling under the counter.

All erections on the poop and bridge, with the exception of the wheel and chart house, are of steel. All bulkheads in living quarters are stopped at the underside of the beams, the space above being fitted with expanded metal panels for ventilation.

Propelling Machinery

The propelling machinery consists of one three-cylinder triple-expansion engine, having cylinders 26½ inches, 45 inches and 75 inches diameter by 48-inch stroke, with independent condenser and bilge pumps. The engine drives a right-hand, four-bladed, built-up propeller with bronze blades and cast iron hub.

The main engines are fitted with Stephenson link motion for valve gear, the high-pressure and intermediate-pressure valves being of the piston type, and the low-pressure of the balanced slide type. The piston rods and valve stems are fitted with Tucker Improved United States double-block metallic packing.

The shafting is 10 percent heavier than Lloyd's requirements with male and female couplings and parallel coupling

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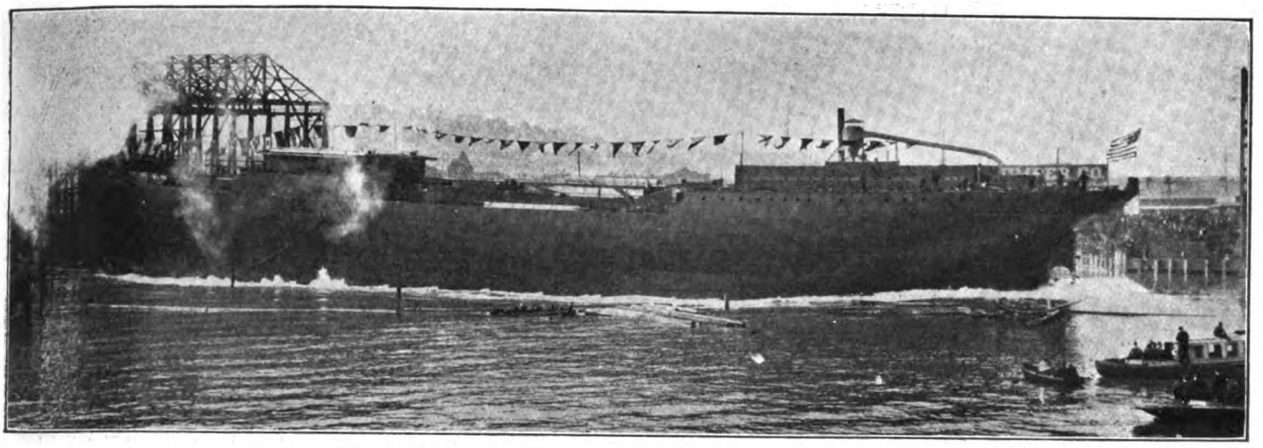

Fig. 1.—Launch of the *Frank H. Buck* at the Union Iron Works, San Francisco, Cal., February 11

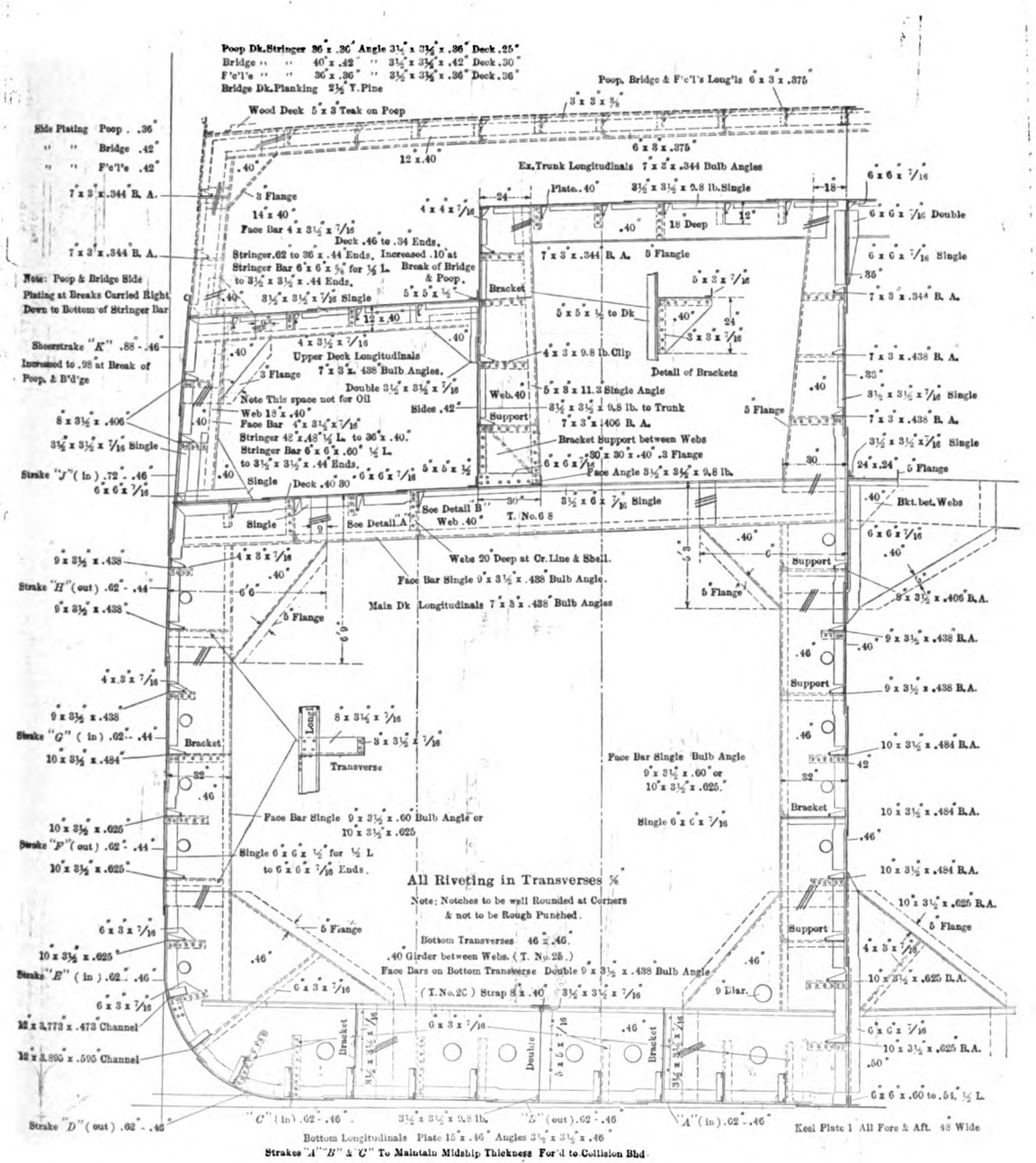
Poop Dk.Stringer 36" x .36" Angle 3½" x 3½" x .36" Deck .25"
Bridge " " 40" x .42" " 3½" x 3½" x .42" Deck .30"
F'c'l'e " " 36" x .36" " 3½" x 3½" x .36" Deck .36"
Bridge Dk.Planking 2½" Y.Pine
Wood Deck 5" x 3" Teak on Poop
Poop, Bridge & F'c'l's Long'ls 6" x 3" x .375"
Side Plating Poop . .36"
" " Bridge .42"
" " F'c'l'e .42"
7" x 3" x .344" B. A.
Note: Poop & Bridge Side Plating at Breaks Carried Right Down to Bottom of Stringer Bar
Shearstrake "A" .38 - .46
Increased to .98 at Break of Poop, & B'd'ge
Face Bar 4" x 3½" x 7/16"
Deck .46 to .34 Ends,
Stringer .62 to .36" x .44" Ends, Increased .10" at Break of Bridge & Poop.
Stringer Bar 6" x 6" x ⅝" for ½ L. to 3½" x 3½" x .44" Ends,
3½" x 3½" x 7/16" Single
Upper Deck Longitudinals 7" x 3" x .438" Bulb Angles,
Double 3½" x 3½" x 7/16"
Note This space not for Oil
Web 18" x .40"
Face Bar 4" x 3½" x 7/16"
Stringer 42" x .48" ½ L. to 36" x .40"
Stringer Bar 6" x 6" x .60" ½ L. to 3½" x 3½" x .44" Ends,
Single Deck .40
Main Dk Longitudinals 7" x 3" x .438" Bulb Angles
Webs 20" Deep at Cr. Line & Shell.
Face Bar Single 9" x 3½" x .438" Bulb Angle.
Strake "H" (out) .62" - .44"
Strake "G" (in) .62" - .44"
Strake "F" (out) .62" - .44"
Strake "E" (in) .62" - .46"
Strake "D" (out) .62" - .46"
Bracket
Transverse
Face Bar Single 9" x 3½" x .60 Bulb Angle or 10" x 3½" x .625
Single 6" x 6" x ½" for ½ L to 6" x 6" x 7/16" Ends.
All Riveting in Transverses ⅞"
Note: Notches to be well Rounded at Corners & not to be Rough Punched.
Bottom Transverses 46" x .46"
.40" Girder between Webs. (T. No. 25.)
Face Bars on Bottom Transverse Double 9" x 3½" x .438" Bulb Angle
(T. No. 26) Strap 8" x .40" 3½" x 3½" x 7/16"
12" x 3.773" x .473" Channel
12" x 3.899" x .595" Channel
Ex. Trunk Longitudinals 7" x 3" x .344" Bulb Angles
Plate .40"
3½" x 3½" x 9.8 lb. Single
Deep
Bracket
5" x 5" x ½" to Dk
3" x 3" x 7/16"
4" x 3" x 9.8 lb. Clip
Detail of Brackets
Web .40"
5" x 3" x 11.3 Single Angle
3½" x 3½" x 9.8 lb. to Trunk
7" x 3" x .406" B. A.
Bracket Support between Webs
30" x 30" x .40" .3 Flange
Face Angle 3½" x 3½" x 9.8 lb.
See Detail B"
See Detail A"
T. No. 6 8
Support
Bkt. bet. Webs
Face Bar Single Bulb Angle 9" x 3½" x .60" or 10" x 3½" x .625"
Single 6" x 6" x 7/16"
Bottom Longitudinals Plate 15" x .46" Angles 3½" x 3½" x .46"
"C" (in) .62" - .46"
3½" x 3½" x 9.8 lb.
"B" (out) .62" - .46"
"A" (in) .62" - .46"
Keel Plate 1 All Fore & Aft. 48" Wide
Strakes "A" "B" & "C" To Maintain Midship Thickness For'd to Collision Bhd.

Fig. 2.—Midship Section of *Frank H. Buck*

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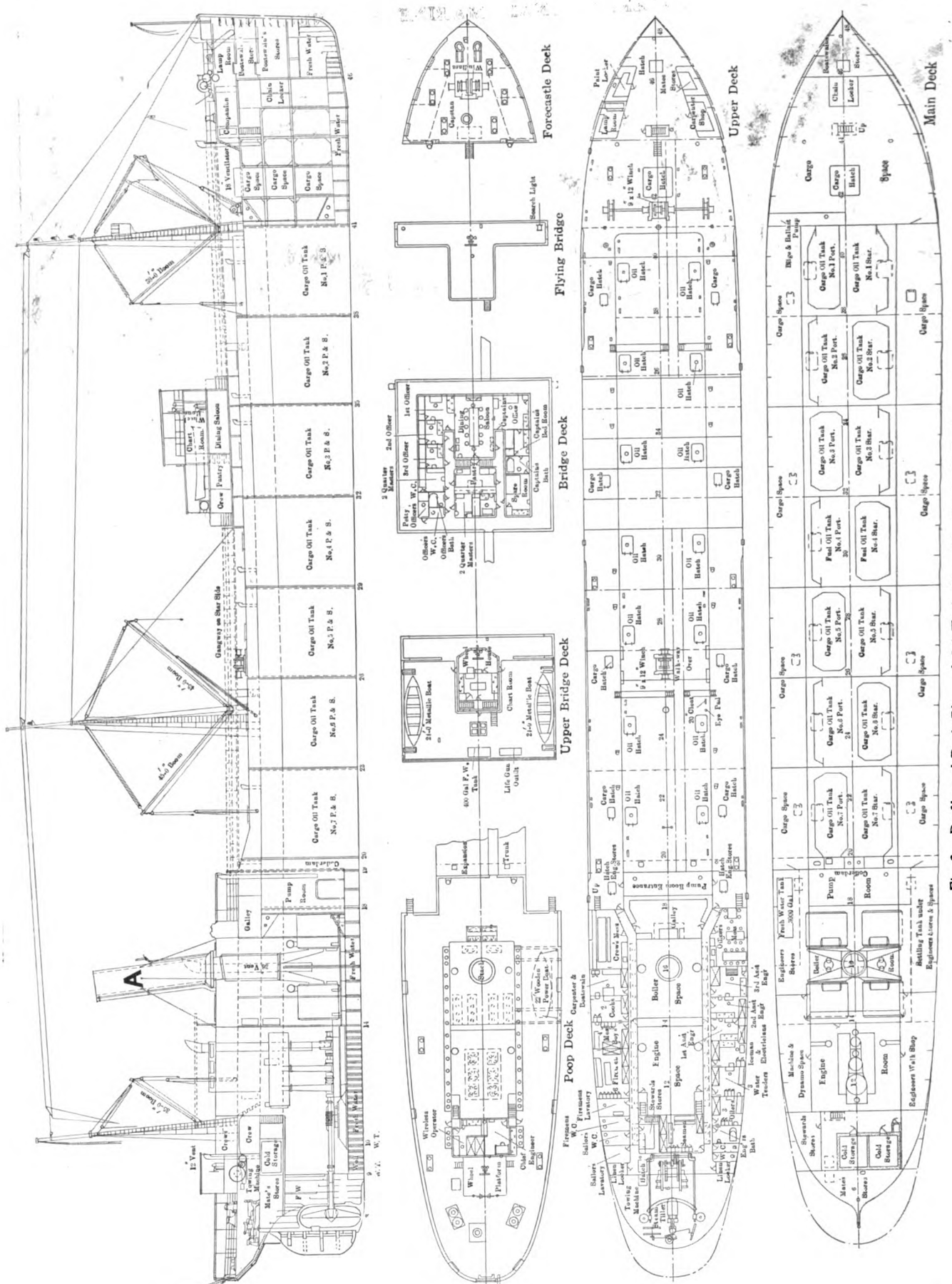

Fig. 3.—Profile and Deck Plans of Oil Tank Steamship *Frank H. Buck*

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bolts. The thrust is of the horseshoe type, with the horseshoes cored for water circulation.

Steam is supplied to the engine by four single-ended Scotch marine boilers, each 14 feet diameter by 12 feet long, built for a steam pressure of 200 pounds per square inch. Each boiler is fitted with three Morison suspension furnaces about 42 inches diameter, with a separate combustion chamber to each furnace. The tubes are 3 inches external diameter, and a steam drum, 3 feet diameter by 6 feet long, is fitted to each boiler. The boilers have a total heating surface of 9,525 square feet and are fitted to burn fuel oil, the oil fuel system being of the Dahl type, which is almost universally used on the Pacific Coast and has given extremely satisfactory results.

An engineer's workshop is fitted up with one 22-inch drill press, one 20-inch by 14-foot lathe, one double emery grinder, bench vise and all necessary tools, the machinery being driven by a 7½ horsepower motor.

Auxiliaries

The auxiliary machinery consists of one main centrifugal circulating pump, driven by an independent direct-acting single-cylinder engine; an independent air pump; an auxiliary air pump: main and auxiliary feed pumps; a fire pump; sanitary pumps, and a feed water heater and evaporator plant.

The steering gear is of the Brown steam tiller type, with telemotor control. The electric plant consists of two 10-kilowatt generators, a switchboard, one 14-inch searchlight, and all accessories. The refrigerating plant is of Vulcan Iron Works' make, of two tons capacity, on the direct-expansion system, connected to the refrigerating chambers and to cooler tanks.

Lifeboat accommodations are provided for all on board the vessel by means of two 24-foot metallic lifeboats fitted complete with all necessary equipment as required by the United States laws. The boats are handled with Welin davits on the upper bridge. One 22-foot power boat is also provided. Life preservers and life buoys are carried as called for by law.

Cargo Handling

The cargo system consists of two duplex pumps having 18-inch cylinders, 15-inch oil cylinders and 18-inch stroke. The suction system consists of two 12-inch mains run one on each side of centerline bulkhead, with a 10-inch branch to each tank. Bypass arrangements are made so that any tank on one side of the ship can be emptied and discharged either overboard, through sea cocks or into any other tank on the opposite side.

Each pump can separately or together discharge into an 8-inch belt discharge main running along the top of the expansion trunk, from which 8-inch branches are fitted for discharging overboard, or back to the tanks by 6-inch branches. Discharges are so arranged that either pump can discharge into one side of the main or the other, and division valves are provided so that one pump can be working at a heavier pressure than the other. An independent 6-inch suction is fitted to one set of the main cargo tanks to discharge into the fuel tanks. The discharge system is so arranged that it can be used as suction for one or both pumps.

Each pair of tanks is fitted with a 6-inch equalizing valve. Two turbine fans are fitted in the pump room to discharge air either into the pump room or into 12-inch suction pipes to the cargo tanks.

Liquid Fuel.—At a meeting of the Institute of Marine Engineers on March 9, a paper was read by Mr. J. Veitch Wilson on "Liquid Fuel." In the course of an interesting historical retrospect, including a personal experience of over fifty years, he said it seemed reasonable to suppose, in view of the fact that so much petroleum had been found by spontaneous exudations and accidental discoveries, that the world was saturated with oil, and that, by systematic investigations, supplies were likely to be discovered in every habitable quarter of the globe. The world's production of crude petroleum showed a steady advance each year from 264,249,119 barrels in 1907 to 351,178,236 barrels in 1912, and he understood that the figures for 1913 showed an advance of 10 percent over those of 1912.

Comparative Costs of Warships Built in Government Navy Yards and in Private Shipyards

In the discussion of the Naval Appropriation Bill before Congress on April 21, Mr. William A. Jones, of Virginia, gave some interesting figures regarding the relative costs of naval vessels built in Government navy yards, and by contract in private shipbuilding establishments. The figures for the most part were supplied either by the Secretary of the Navy or by the Chief of the Bureau of Construction and Repair, and include, besides the first cost of such vessels, a record of the cost of repairs on the ships since they have been in service. Taking the case of the battleship *Connecticut*, which was built eight years ago in the New York navy yard, it was shown that the *Connecticut* cost $374,000 (£76,600) more than a sister ship, the *Louisiana*, built by contract at the yards of the Newport News Shipbuilding & Dry Dock Company, Newport News, Va., the cost of repairs on the *Connecticut* up to March 1, 1914, amounting to $31,000 (£6,360) more than the cost of repairs on the *Louisiana* up to the same date. Again, in the case of the Florida, which was also built at the New York navy yard, the first cost of the ship was $2,269,000 (£465,000) more than that of the *Utah*, a sister ship, built by contract at the New York Shipbuilding Company, Camden, N. J. The cost of repairs on the *Florida* up to March 1, 1914, amounted to $55,000 (£11,250) more than the cost of repairs on the *Utah*. Still another example for comparison was offered by the completion this year of the battleships *New York* and *Texas*. The *New York* was built in the New York navy yard, and cost $1,463,000 (£300,000) more than the *Texas*, a sister ship built by the Newport News Shipbuilding & Dry Dock Company, Newport News, Va. Notwithstanding that the total expenditures on the battleship *New York* up to March 1, 1914, indicate savings of $413,750.50 (£84,700) for the construction of the hull and $245,884.91 (£50,400) for the construction of the machinery as compared with the original estimates for the ship, and that additional amounts above the contract price of the *Texas* were paid by the Government for extra work on the *Texas*, nevertheless the Government-built ship *New York* cost the Government more than a million dollars more than the contract-built ship *Texas*. At present the Navy Department is requesting an appropriation for the construction of two battleships, eight destroyers and three submarines, the cost of which, exclusive of armor and armament, according to estimates of the Navy Department, would be, if built in private yards, about $25,600,000 (£5,250,000), while if the ships were built in navy yards the cost would be about $32,003,000 (£6,565,000), or about 28 percent more than the cost under private contract. The significance of these figures led Mr. Jones to oppose very strongly the policy of the Government in placing large contracts for the construction of naval vessels with navy yards, only two of which are at present properly equipped for building large ships, whereas there are now in the United States no less than five large private shipyards which have been developed in recent years very largely for the purpose of undertaking such work, and which have amply proved their ability to turn out the high grade of work demanded by the Navy Department under the most rigid inspection at a much less cost than is possible in navy yards.

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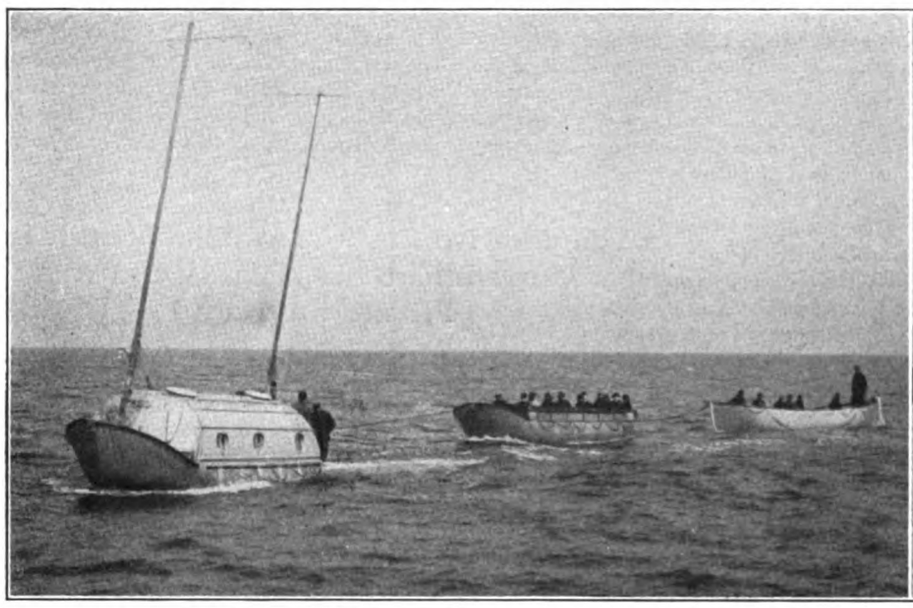

Fig. 1.—Lundin Power Lifeboat Towing Open Lifeboats

Fig. 2.—Open Power Lifeboat, with 64 Persons on Board

Tests of Lundin Lifeboats

The type of lifeboat developed by Capt. A. P. Lundin, president of the Welin Marine Equipment Company, Long Island City, N. Y., has been described in previous issues of this journal, and its principal features are probably well known to most of our readers. The Lundin lifeboat is essentially a broad, flat-bottom, round-ended scow-shaped boat built of high-grade steel plates galvanized as soon as they are rolled so that they are thoroughly protected from corrosion. The boats are completely decked over, with the space between the steel deck and the bottom of the boat subdivided by seven watertight bulkheads into watertight compartments, each compartment being fitted with a watertight manhole plate and self-acting scuppers, provided to carrying off any water that might get into the boat. The shape of the boat itself is conducive to strength, being that of a steel box girder of sufficient strength to stand the strain of being hoisted by steamships' davits with a full load on board.

Above the steel deck numerous heavy 'thwarts cross the boat, stiffening the hull still further and making her capable of withstanding the shocks to which she might be subjected in swinging against a ship's side while being lowered. As a further precaution, a thick Balsa wood fender, firmly fastened, but, at the same time, removable to permit of painting the inner surface, is fitted to each side of the boat. This fender not only rounds out the boat's shape nicely and protects it from damage, but also adds to its stability, as Balsa wood is much lighter than cork.

In the ordinary open Lundin lifeboats the sides and ends above the 'thwarts are made of wood capable of folding down to allow nesting of the boats one above the other, or, if it is unnecessary to nest the boats, the top sides can be made stationary.

One of the most recent types of lifeboats designed by Capt. Lundin is the power lifeboat illustrated in the accompanying photographs. A complete description of this type of boat was published in our June, 1912, issue, but since that date some new features have been incorporated in the design.

The power boat is practically a development of the standard Lundin decked lifeboat, but it is provided with a tunnel

Fig. 3.—Housed-in Power Lifeboat, with 50 Persons Inside and 17 Outside

Fig. 4.—28-Foot Lundin Lifeboat, with 61 Persons on Board, Lowered to the Water by Welin Quadrant Davits

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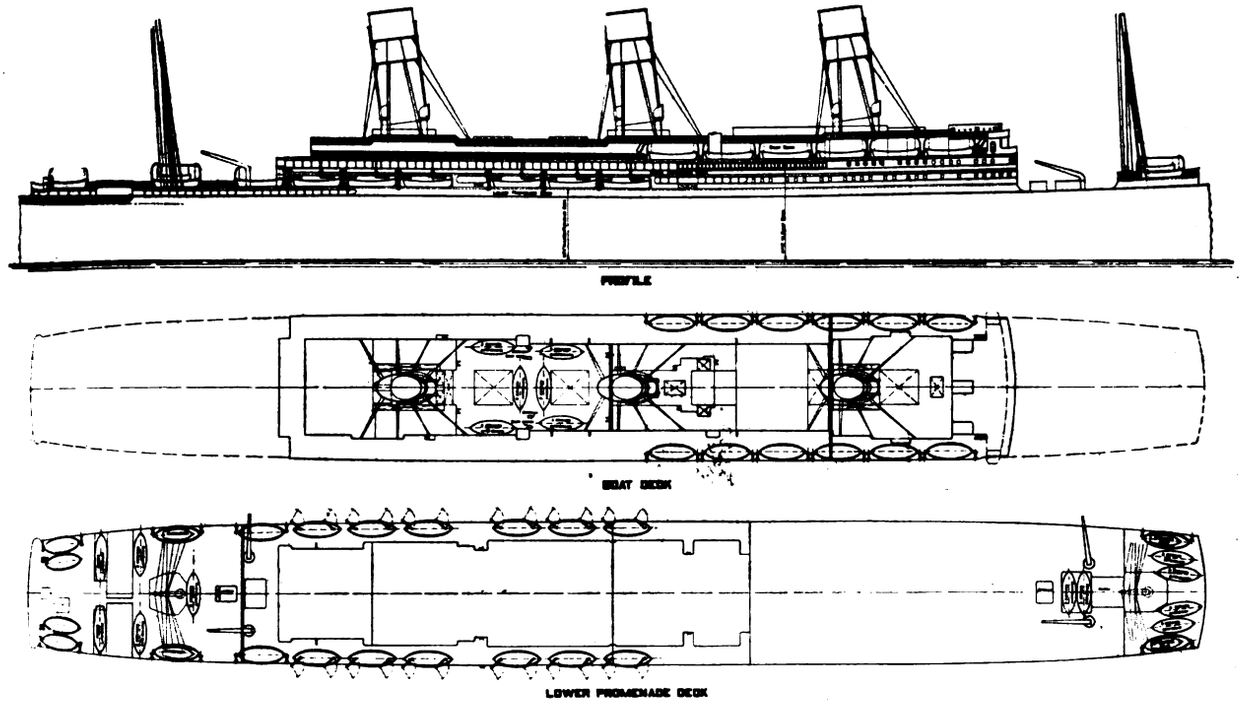

Fig. 5.—Present Arrangement of Lifeboats on the Hamburg-American Liner *Imperator*

stern in which the propeller is placed, so that it is entirely above the bottom of the boat, and well protected. The engine is placed in a watertight metal compartment, a little aft of amidships, so arranged that the lid forms a seat with the spark, throttle and clutch levers all arranged handy to the man at the helm, who thus after the motor has been started has full control of the boat.

The power lifeboat is fitted with tall hinged masts for carrying a wireless aerial, while the wireless apparatus is installed inside a Balsa wood silence cabin built in the forward part of the boat on the starboard side. The wireless equipment is a complete Marconi outfit similar to that used in the United States submarines for sending and receiving messages at distances from 50 to 100 miles at sea. Reels of strong rope and line-carrying guns for establishing communication with other boats, or to a ship, for the purpose of towing or rescue work, are also provided.

In view of the fact that many shipwrecked persons die of exposure in open boats, and that many women and children are included in the passenger lists of to-day, Capt. Lundin has had a strong but light sheet steel cabin put over these boats, and they are now being tested out. The steel cabin is built with large watertight doors at either end, and combination watertight port lights and rowlocks fitted along each side, enabling the boat to be maneuvered with oars while getting clear of the ship. This arrangement has proved a success

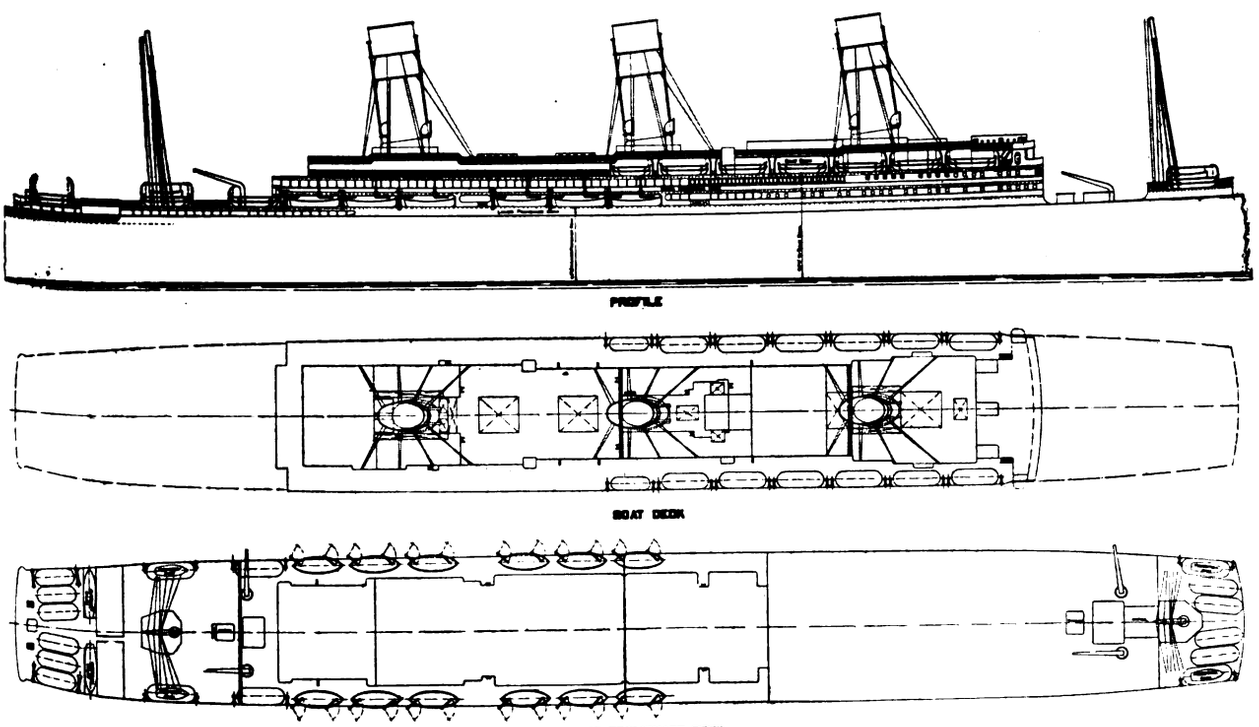

Fig. 6.—Proposed Arrangement of Lifeboats on the *Imperator* with Lundin Type of Boats

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and involves no hindrance to the handling of the boat with oars. In extremely bad weather the windows and doors of the cabin can be clamped down absolutely watertight, as was proved under test when a 24-foot boat fitted with a similar cabin was parbuckled upside down, and as soon as the rope was slacked the boat righted herself immediately without shipping any water.

Recently a series of interesting tests of the Lundin lifeboats was made in the presence of General Uhler, Supervising Inspector-General of the United States Steamboat Inspection Service, and eight members of the Steamboat Inspection Service. In testing a 30-foot by 10-foot steel Lundin housed power lifeboat, fifty men were ordered by the inspectors to be seated inside the cabin, and when the boat was thus loaded

Fig. 7.—Interior of Cabin on Lundin Power Lifeboat, Showing Wireless Room and Motor

seventeen more men stood on the edge of the boat, outside the cabin, listing her only slightly, as shown in Fig. 3, her guard rail still being 2 inches above the water. It is difficult to say how many persons such a boat could save in case of disaster at sea, for, besides the fifty which could easily be accommodated inside the cabin, many more could in an emergency crowd onto her decks, hang onto the cabin top or to the lifeline stretched in loops along her sides.

In testing the seagoing qualities of the power lifeboat before the steamboat inspectors, the four-cylinder, 24 to 30-horsepower Standard motor, turning a 22-inch diameter three-bladed propeller, operating in the tunnel of the boat, gave the boat a speed of 6½ miles an hour, while taking a couple of other lifeboats in tow made no material difference in the speed. It is evident that such a boat could tow a string of other lifeboats and hold them head on to the sea, while by means of her wireless equipment calls could be sent to passing steamships.

The fact that the Lundin boats can be nested, so that two or more boats can be handled under a single set of davits, gives an opportunity for improving the conditions of stowing lifeboats on board ship. In the case of the largest passenger steamship now in service, the Hamburg-American Liner *Imperator*, the present arrangement of the lifeboats provides accommodations in lifeboats under davits on the boat, lower promenade decks, and on the forward and after deck houses for only 2,300 persons, while accommodations for 2,953 others are provided by lifeboats handled under booms. By using Lundin lifeboats, which can be nested two under each set of davits, accommodations for 5,160 persons can be provided by lifeboats under davits, as shown in the proposed arrangement indicated in Fig. 6. Only two lifeboats are handled under booms, these being motor boats with a total capacity for only 160 persons.

Chinese Junk Ning-Po

Few ships of any type now afloat can compare in age with the Chinese junk *Ning-Po*, a famous old smuggler and privateer craft now lying in Los Angeles harbor. Over a year ago this remarkable vessel was navigated from the Chinese coast to California under a Chinese crew and a Danish master, laden with a cargo of Chinese curios for exhibition at the Panama-Pacific Exposition next year.

Built so long ago as 1806, it was not without difficulty that the voyage of 7,000 miles across the North Pacific was accomplished. Off the coast of Japan she was swept by a vicious typhoon, and was partly dismantled by the storm. Part of her Chinese crew was lost through mutiny, and she suffered the humiliation of being rescued by a Japanese cruiser. After having weathered the gales of one hundred years, however, the barnacled bottom, weakened timbers and generally weather-worn appearance of the vessel are no rebuke to her remarkable Oriental builders.

The *Ning-Po* left the port of Shanghai September 17, 1912, and arrived in the harbor at San Pedro, Cal., February 10, 1913, being the first vessel to enter the port flying the flag of the new Chinese republic. For almost a year she laid at the various coast resorts adjacent to Los Angeles, and was finally moored in the harbor under the lee of the Government breakwater. One day last fall, as she swung unmanned at her moorings behind the protecting breakwater, a heavy storm overwhelmed her, and, with as little dignity as a mud scow, the vessel deliberately went to the bottom, only her masts remaining visible.

After this mishap, new owners acquired the *Ning-Po* and she was raised from the mud and taken to the Craig shipyard, Long Beach, Cal., for overhauling. Upon a recent visit to this yard, Mr. William T. Donnelly, consulting engineer, New York, had an opportunity of examining the vessel while she was in dry dock, and to him we are indebted for the following information.

The *Ning-Po* is 150 feet long, 35 feet beam, with a depth of but 15 feet. She has three decks, nine watertight bulkheads, and a maze of secret compartments that only a Chinese mind could conceive, making it apparent that she was built primarily for the smuggling trade. In dark corners in her hold are series of hiding places, many of her timbers being "gophered" out and the openings artfully concealed.

The vessel is of remarkable and unique construction, and to the Eastern shipbuilder most resembles the "buck-eye," familiar to the waters of Chesapeake Bay. The planking, which is of heavy timbers, is built up and bolted edgeways. Three heavy bilge strakes are from half-round logs scarfed to provide the curvature at the stern and bow. The vessel is calked both inside and out, and after being practically completed as to form and planking the frames are added from the inside.

The very strange and unique bow has quite a pleasing appearance, as it is decorated with the bulging eyes of a giant dolphin. This is a common feature of all Oriental vessels, as the superstitious Chinese sailor of the early days would as soon think of building a ship without a rudder as without an eye to see the dangers ahead.

The curious high stern resembles the historical vessels of the Columbus fleet. The planking of the stern is recessed, and this recess has a step to limit the set of the rudder, which, by the way, is no toy, as it is far heavier than would be expected on a vessel of this size. Across the upper part of the stern a counterboard, highly decorated, is worked in, and somewhere on this board is the name of *Ning-Po*. One of the illustrations shows an elaborate dragon worked on the starboard quarter.

Altogether the structure shows a surprising degree of skill and thoroughness in building, as evidenced by the durability of her camphorwood hull and her ironwood masts. Fig. 2 shows the rig to be fore-and-aft or a schooner rig with a

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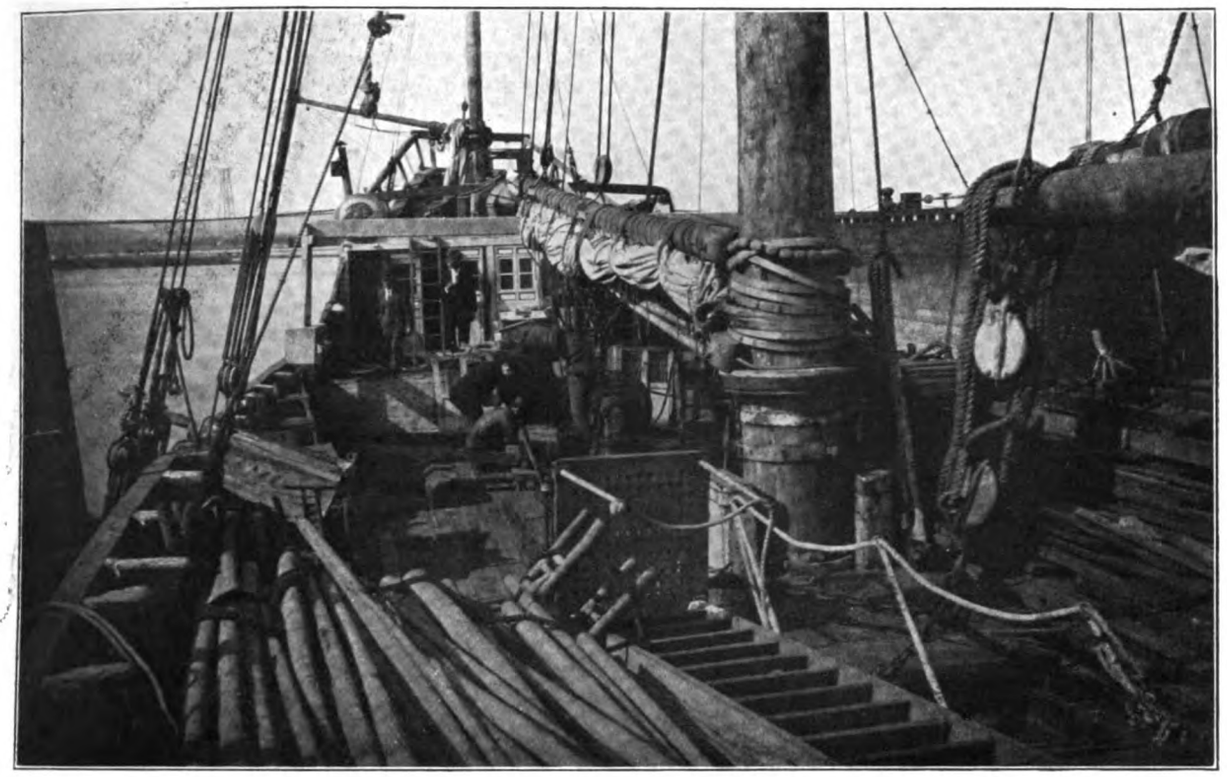

Fig. 1.—View on Deck of *Ning-Po*, looking aft

jigger mast on the high poop. This is the only apparent modification from true Oriental lines. A very novel feature observed was a hawser made from split and twisted bamboo. This was about 2½ inches in diameter and looked to be very strong. The rigging is of the familiar manila rope.

The *Ning-Po* was built at Foo-Chow in 1806, supposedly for the regular Formosan trade, although her intricate interior indicates that she was actually built for the popular and remunerative trade in which she was engaged through most of her sea-faring life, that of smuggling opium and slave girls. The original name of the *Ning-Po* was *Kin Tai Foong*, but in 1861 she was confiscated by the *Chinese Gordon* for rendering aid to the rebels at Ning-Po and her name was changed to the one which she has retained since.

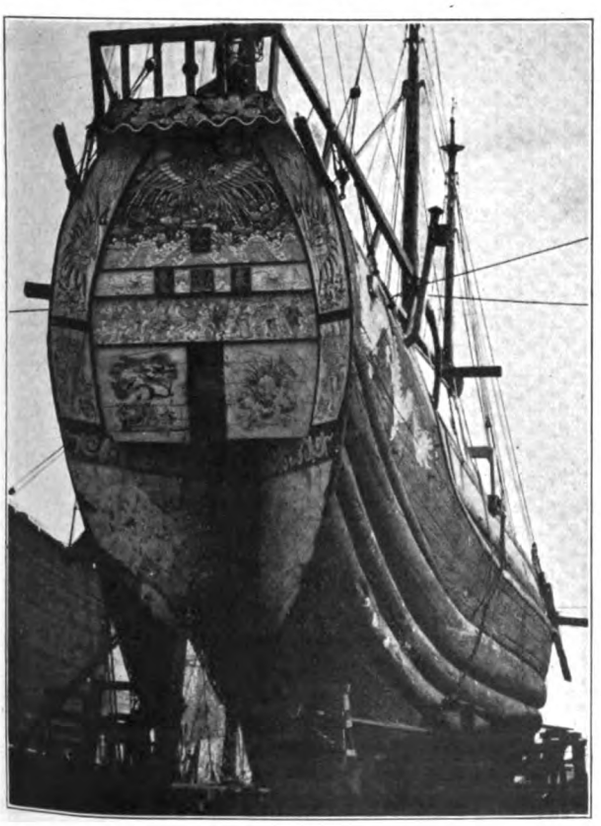

Fig. 2.—Stern of *Ning-Po*

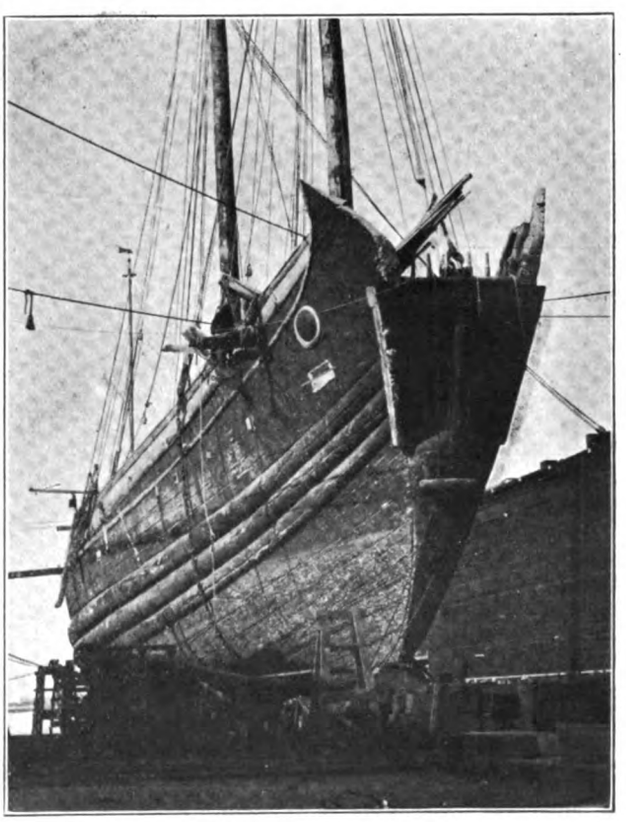

Fig. 3.—The *Ning-Po* in Drydock, Showing Heavy Bilge Strakes

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Convention for the Safety of Life at Sea

Abstract of the Text and Regulations of the International Convention for the Safety of Life at Sea Signed at London, January 20

The provisions of this convention are completed by regulations which have the same force and take effect at the same time as the convention. Every reference to the convention implies at the same time a reference to the regulations, copies of which should be secured for a complete understanding of the convention.

Except where otherwise provided by this convention, the merchant vessels of any of the contracting States which are mechanically propelled, which carry more than twelve passengers, and which proceed from a port of one of the said States to a port situated outside that State, or conversely, are subject to the provisions of this convention. Ports situated in the colonies, possessions or protectorates are considered to be ports outside the contracting States.

Persons who are on board by reason of force majeure or in consequence of the obligation laid upon the master to carry shipwrecked or other persons are not deemed to be passengers.

There are excepted from this convention, save in the cases where the convention otherwise provides, vessels making voyages specified in a schedule to be communicated by each high contracting party to the British Government at the time of ratifying the convention.

No schedule may include voyages in the course of which the vessels go more than 200 sea miles from the nearest coast.

Each high contracting party has the right subsequently to modify its schedule of voyages in conformity with this article on condition that it notifies the British Government of such modification.

Each high contracting party has the right to claim from another contracting party the benefit of the privileges of the convention for all of its vessels which are engaged in any one of the voyages mentioned in its own schedule. For this purpose the party claiming such benefit shall impose on the said vessels the obligations prescribed by the convention in so far as, having regard to the nature of the voyage, these obligations would not be unnecessary or unreasonable.

No vessel, not subject to the provisions of the convention at the time of its departure, can be subjected to the convention in the course of its voyage, if stress of weather or any other cause of force majeure compels it to take refuge in a port of one of the contracting States.

SAFETY OF NAVIGATION

When the expression "every vessel" is used forthwith, it includes all merchant vessels which belong to any of the contracting States.

The contracting parties undertake to take all steps to insure the destruction of derelicts in the northern part of the Atlantic Ocean east of a line drawn from Cape Sable to a point situated in latitude 34 degrees North and longitude 70 degrees West. Further, they will establish in the North Atlantic with the least possible delay a service for the study and observation of ice conditions and a service of ice patrol. For this purpose

Two vessels shall be charged with these three services.

During the whole of the ice season, they shall be employed in ice patrol.

During the rest of the year the two vessels shall be employed in the study and observation of ice conditions and in the destruction of derelicts; nevertheless the study and observation of ice conditions shall be effectively maintained, in particular from the beginning of February to the opening of the ice season.

While the two vessels are employed in ice patrol the contracting parties, to the extent of their ability and so far as the exigencies of the naval service will permit, will send warships or other vessels to destroy any dangerous derelicts, if this destruction is considered necessary at that time.

The government of the United States is invited to undertake the management of the three services of derelict destruction, study and observation of ice conditions, and ice patrol. The contracting States which are specially interested in these services undertake to contribute to the expense of establishing and working the said services in the following proportions: United States of America, 15 percent; France, 15 percent; Germany, 15 percent; Great Britain, 30 percent, and the others in amounts varying from 2 to 4 percent.

The master of every vessel which meets with dangerous ice or a dangerous derelict is bound to communicate the information by all the means of communication at his disposal to the vessels in the vicinity, and also to the competent authorities at the first point of the coast with which he can communicate. Every administration which receives intelligence of dangerous ice or a dangerous derelict shall take all steps which it thinks necessary for bringing the information to the knowledge of those concerned and for communicating it to other administrations. The transmission of messages respecting ice and derelicts is free of cost to the vessels concerned. It is desirable that the said information should be sent in a uniform manner. For this purpose, a code, the use of which is optional, has been prepared.

The master of every vessel fitted with a radiotelegraph installation, on becoming aware of the existence of an imminent and serious danger to navigation, shall report it immediately. When ice is reported on, or near, his course, the master of every vessel is bound to proceed at night at a moderate speed, or to alter his course so as to go well clear of the danger zone.

The use of the international distress signals for any other purpose than that of signals of distress is prohibited on every vessel. The use of private signals which are liable to be mistaken for the international distress signals is prohibited on every vessel.

The selection of the routes across the North Atlantic in both directions is left to the responsibility of the steamship companies. Nevertheless, the contracting parties undertake to impose on these companies the obligation to give public notice of the regular routes which they propose their vessels should follow, and of any changes which they make in them. They undertake, further, to use their influence to induce the owners of all vessels crossing the Atlantic to follow as far as possible the routes adopted by the principal companies.

The contracting parties undertake to use all diligence to obtain from the governments which are not parties to this convention their agreement to the revision of the International Regulations for Preventing Collisions at Sea, as indicated below:

Article 2. The second white mast-head light to be compulsory. Article 10. A permanent fixed stern light to be compulsory. Article 14. A special day signal to be compulsory for motor vessels. Article 15. A special sound signal to be established for use by a vessel in tow, or if the tow is composed of several vessels, by the last vessel of the tow. Ar-

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ticle 31. Add to the lists of both day and night signals the international radiotelegraph distress signal.

The governments of the contracting parties undertake to maintain, or, if it is necessary, to adopt, measures for the purpose of insuring that, from the point of view of safety of life at sea, the vessels to which this convention applies shall be sufficiently and efficiently manned.

CONSTRUCTION

"New vessels" are those the keel of which is laid after July 1, 1915. Other vessels are considered as "existing vessels." Existing arrangements on each of these vessels shall be considered by the administration of the State to which the vessel belongs, with a view to improvements providing increased safety where practicable and reasonable.

SUBDIVISION OF VESSELS

Vessels shall be as efficiently subdivided as is possible, having regard to the nature of the service for which they are intended. The minimum requirements respecting subdivision and arrangements affecting subdivision are given. The degree of safety provided for by these minimum requirements varies in a regular and continuous manner with the length of the vessel and with a certain "criterion of service." The requirements of the annexed regulations are such that the highest degree of safety corresponds with the vessels of greatest length primarily engaged in the transportation of passengers. The regulations indicate the method to be followed in order to determine the permissible length of compartments on the basis of the floodable length; prescribe a limit to the length of compartments; and fix the conditions governing certain special cases.

When the watertight subdivision of a vessel is such as to provide for a degree of safety greater than that provided by the rules prescribed by this convention, the administration of the State to which the vessel belongs shall, if so requested by the owner, record this fact on the safety certificate of the vessel.

FLOODABLE LENGTH

The floodable length at any point of the length of a vessel shall be determined taking into consideration form, draft, and other limiting characteristics of the vessel in question.

This floodable length for a given point in a vessel with a continuous bulkhead deck is the maximum percentage of the length of the vessel (having its center at the point in question) which can be flooded under the definite assumptions hereafter set forth without the vessel being submerged beyond the margin line.

In the case of vessels not having a continuous bulkhead deck, the floodable length must be such as to secure to the vessel in question, for each portion of its length, and for all conditions of trim after damage, a measure of safety at least equal in effectiveness to that laid down for the vessel with continuous bulkhead deck.

In determining the floodable length a uniform average permeability shall be used throughout the whole length of each of the three following portions of the vessel:

(1) The machinery space;
(2) The portion forward of the machinery space; and
(3) The portion abaft the machinery space.

For steam vessels the permeability of the machinery space, including the double bottom in wake thereof, shall be taken as 80 percent. For vessels fitted with internal combustion engines the corresponding permeability shall be taken as 85 percent, unless it is proved by actual calculation that a lower figure may be adopted, provided that in no case shall that figure be less than 80 percent.

The permeabilities for spaces forward and aft of the machinery space shall be as follows:

(*a*) Sixty percent in cargo spaces, bunkers (permanent or reserve), storerooms, baggage and mail rooms, chain lockers, watertight shaft or pipe tunnels, and fresh-water tanks above the double bottom.

It must be proved that the spaces just enumerated are practicable for the purpose intended and that they are in fact to be so used. The same permeability shall not be assigned to any other space without the approval of the administration.

(b) Ninety-five percent in passenger and crew spaces, peaks, trimming tanks exclusively so used, double bottoms, and all other spaces not specifically appropriated to one of the purposes indicated in the foregoing section (*a*).

If in a 'tween deck space inclosed by complete transverse permanent steel bulkheads any portion thereof is appropriated to passengers, the whole of that space shall be regarded as passenger space; and, similarly, 'tween deck spaces appropriated for the carriage of *either* passengers *or* cargo shall be regarded as passenger spaces.

Where the spaces before or abaft the machinery space below the margin line consist partly of spaces mentioned in section (*a*) and partly of spaces mentioned in section (*b*), the average percentage of permeability shall be determined separately for each end by the formula $95-35r$, where r is the ratio between the volume of the spaces mentioned in section (*a*) and the total volume of the space in the portion of the ship under consideration.

PERMISSIBLE LENGTH OF COMPARTMENTS

(1) The maximum permissible length of one compartment having its center at any point in the vessel's length is obtained from the floodable length by multiplying that length by an appropriate factor, called the *factor of subdivision*.

(2) This factor of subdivision depends on the length of the vessel, and, for a given length, varies according to the nature of the service for which the vessel is intended. This factor decreases in a regular and continuous manner—

(*a*) As the length of vessel increases; and

(*b*) As, for a given length, the vessel departs from the type of vessel engaged in a mixed cargo and passenger service, and approaches to the type of vessel primarily engaged in the transportation of passengers.

(3) For each of the two types of vessels referred to in the previous paragraph (2) (*b*) the variation of the factor of subdivision may be expressed by a curve, of which the coordinates represent the length of the vessel and the value of the factor. The following table gives certain points on two curves the higher of which corresponds to the minimum requirements for the "mixed" type, and the lower to the minimum requirements for the "passenger" type.

TABLE

A.	B. Meters.	B. Feet.	C. Meters.	C. Feet.
1.00	90	295	79	259
0.90	114	374	87	285
0.84	123	404	93	305
0.65	149	489	116	380
0.50	174	571	149	489
0.39	213	699	209	685
0.34	274	899	274	899

Column A gives the maximum permissible values of the factor of subdivision for the length of vessels given in Columns B and C.

Column B is applicable to vessels engaged in the mixed cargo and passenger service.

Column C is applicable to vessels primarily engaged in the transportation of passengers.

(4) For a given length, the value of the factor of subdi-

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vision appropiate to a vessel between the two extreme limits will be between the values of the factors determined by the two curves before mentioned, and will be automatically fixed by a "*criterion of service*" which is to form the subject of further study.

(1) When the factor of subdivision is equal to or less than .5, it may be doubled in order to give at any point of the vessel's length the total length of two adjacent compartments; but in that case the length of the shorter compartment of any pair shall not be less than one-quarter of the total length so obtained. If one of the two adjacent compartments is situated inside the machinery space and the second is situated outside the machinery space, and the average permeability of the portion of the ship in which the second is situated differs from 80 percent, the length of the pair of compartments shall be adjusted to the proper value by applying a suitable correction.

(2) In no case whatever shall the length of any watertight compartment exceed 28 meters (equivalent to 92 feet).

(3) When the factor of subdivision applicable to any vessel is less than .84, but more than .5, the combined length of the two foremost compartments shall not exceed the floodable length at the extreme forward end, provided also that the length of the second compartment is not greater than that permissible by the above, and not less than 3 meters (equivalent to 10 feet).

(4) When the length of the vessel is more than 213 meters (equivalent to 699 feet) but less than 251 meters (equivalent to 823 feet) the floodable length at the forward end of the vessel shall be at least 20 percent of the vessel's length; and the vessel, forward of a bulkhead placed either at the distance of the actual floodable length abaft the stem or not nearer to the stem than 20 percent of the vessel's length, shall be divided into at least three compartments.

(5) When the length of the vessel is equal to or greater than 251 meters (equivalent to 823 feet) the same method shall be adopted, but the floodable length shall be at least 28 percent and the number of compartments at least four.

(6) A bulkhead may be recessed transversely, provided the sides of the recess are of a sufficient distance from the sides of the vessel.

Vertical steps are inadmissible in the main transverse watertight bulkheads of vessels to which the subdivision rules apply where the factor of subdivision is greater than .5, unless such arrangements are made by additional subdivision as shall maintain the same measure of safety as that secured by bulkheads without steps. The total length of the steps in any bulkhead shall not exceed 2 percent of the vessel's length, plus 3 meters (equivalent to 10 feet).

(7) The existence of recesses or steps in a bulkhead shall in no case affect the permissible volumes of the compartments adjacent to such bulkhead, as determined by this and the preceding article.

Peak and Machinery Space Bulkheads

Vessels shall be fitted with a forepeak bulkhead to extend to the bulkhead deck, and to the weather deck in vessels having continuous superstructures. This bulkhead shall be placed at a distance of not less than 5 percent of the vessel's length from the stem at the load waterline.

An afterpeak bulkhead and bulkheads dividing the machinery space from the cargo and the passenger spaces shall also be fitted and carried up to the bulkhead deck. The afterpeak bulkhead may, however, be stopped below the bulkhead deck, provided that it shall at least be carried to the first deck above the load waterline, and that such deck forms a watertight flat from the afterpeak bulkhead to the stern, and also provided that the degree of safety of the vessel as regards subdivision is not thereby diminished.

Fireproof Bulkheads

In parts of a vessel above the margin line there shall be fitted fireproof bulkheads which will serve to retard the spread of fire. The mean distance between any two consecutive bulkheads of this description shall not be greater than 40 meters (equivalent to 131 feet). Recesses in these bulkheads shall be fireproof, and the openings in these bulkheads shall be fitted with fireproof doors.

Exits from Watertight Compartments

(1) In passenger and crew spaces a practicable means of escape for the occupants shall be provided from each watertight compartment.

(2) There shall be a means of escape for the crew from each engine room, shaft tunnel and stokehold compartment independent of the watertight doors.

Construction and Initial Testing of Watertight Bulkheads

(1) Watertight bulkheads shall be constructed in such a manner that they shall be capable of supporting, with a proper margin of resistance, the pressure due to a head of water up to the margin line.

(2) Steps and recesses in bulkheads shall be as watertight and as strong as the bulkhead at the place where each occurs.

Where frames or beams pass through a watertight deck or bulkhead, the watertightness shall be obtained by calked angle chocks, or cast iron or steel chocks efficiently secured and rust-jointed, and not by wood or cement.

(3) Testing main compartments by filling them with water is not compulsory. A complete examination of the bulkheads shall be made by a surveyor; and, in addition, a hose test shall be made in all cases.

(4) The foremost and aftermost compartments shall be tested with water to a head up to the margin line.

Double bottoms, double tanks, and all compartments intended to hold liquids shall be tested with water to a head of 2.44 meters (8 feet) above the top of the tank or to the load waterline, whichever is the greater.

(5) No change may be made in the structure of the bulkheads after the completion of the survey without the permission of the administration.

(6) All provisions relating to main transverse watertight bulkheads shall apply to longitudinal bulkheads, so far as is practicable.

Openings in Watertight Bulkheads

(1) The number of openings in watertight bulkheads shall be reduced to the minimum compatible with the design and proper working of the vessel; satisfactory means shall be provided for closing these openings.

(2) No doors, sluice valves, manholes, or access openings are permitted—

(*a*) In the collision bulkhead below the margin line.

(*b*) In watertight transverse bulkheads dividing a cargo space from an adjoining cargo space or from a reserve bunker, except as provided in paragraph 6 of this article.

(3) In the machinery space and apart from bunker and shaft-tunnel doors, not more than one door may be fitted in each main transverse bulkhead within the machinery space for intercommunication, but where more than one separate shaft tunnel is fitted a door may be cut for each tunnel.

If a tunnel is fitted forward either for the purpose of pipes or as a communication passage it shall be fitted with a watertight door.

(4) The only types of watertight doors permissible are hinged doors, sliding doors, and doors of any other equivalent pattern, excluding plate doors secured only by bolts.

A hinged door shall be fitted with lever-operated catches workable from each side of the bulkhead.

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A sliding door may have a horizontal or vertical motion. If hand-operated only, the door shall be capable of being operated at the door itself and also from an accessible position above the margin line. If operated by power, it shall be capable of being operated from the bridge, and by hand at the door itself and from an accessible position above the margin line. A door dropping by its own weight, and fitted with a cataract cylinder or equivalent arrangement, may be considered as being operated by power, if capable of being released from the bridge.

(5) In the case of watertight bunker doors, satisfactory arrangements shall be made by means of screens or otherwise, to prevent the coal from interfering with the closing of the doors.

(6) Hinged watertight doors in passenger, crew, and working spaces are only permitted above a deck the under side of which, at its lowest point at side, is at least 2.13 meters (7 feet) above the load waterline, and they are not permitted in those spaces below such deck.

Hinged watertight doors of specially heavy design may be fitted above the load waterline in bulkheads between cargo 'tween-deck spaces. They shall be closed before the voyage commences, and kept closed while at sea by efficient closing gear. None of these doors shall be fitted, even at the ends of the vessel, in a cargo 'tween-deck space in the amidship region of which 'tween-deck space it would not be permissible to fit such doors.

(7) All other watertight doors shall be sliding doors.

(8) (*a*) When the number of watertight doors in the main transverse watertight bulkheads at or about the stokehold level in the machinery space exceeds five, excluding the watertight doors at the entrances of tunnels, all watertight doors situated below the load waterline shall be capable of being simultaneously closed from a station situated on the bridge, and their opening and closing shall be indicated at that station. The simultaneous closing of these doors shall be preceded by a warning sound signal.

(*b*) If watertight doors which have sometimes to be open at sea for the purpose of trimming coal are fitted between bunkers in the 'tween-decks opening below the bulkhead deck, these shall be operated by power. The opening and closing of these doors shall be recorded in the official log.

(*c*) When trunkways in connection with refrigerated cargo are carried through more than one main transverse watertight bulkhead, and the sills of the openings are less than 2.13 meters (7 feet) above the load waterline, the watertight doors at such openings shall be operated by power.

(9) Portable plates on bulkheads shall not be permitted except in machinery spaces. Such plates shall always be in place before the vessel leaves port, and shall not be removed at sea except in case of urgent necessity. The necessary precautions shall be taken in replacing them to insure that the joint shall be perfectly watertight.

(10) All watertight doors shall be kept closed during navigation except when necessarily opened for the working of the vessel, and shall always be ready to be immediately closed.

(11) If trunkways for forced draft, for access from crew's accommodation to the stokehold or for any other purpose, are carried through the main transverse watertight bulkheads, the integrity of the watertight bulkheads shall be maintained by watertight doors or other equally effective means.

(12) Where pipes, electric light cables, etc., are carried through transverse watertight bulkheads below the margin line, arrangements shall be made to insure the integrity of the watertightness of the bulkheads.

(13) The number of sluice valves in watertight bulkheads shall be reduced to the minimum, and shall not be allowed except in positions where they are sufficiently accessible at all times to allow of its being ascertained that they are in good order. They shall be strongly constructed, efficiently fitted, and regularly inspected. Satisfactory provision shall be made for operating them from an accessible position above the margin line. Means shall be provided for indicating when they are open or shut.

OPENINGS IN VESSEL'S SIDE

(1) (*a*) Subject to clause (*b*) below, when side scuttles are fitted below a deck the under side of which at its lowest point at side is less than 2.13 meters (7 feet) above the load waterline, they shall be permanently fixed.

(*b*) Side scuttles which are capable of being opened may be fitted in the positions defined in clause (*a*), provided that—

They shall be closed watertight and locked before the vessel leaves port; they shall not be opened during navigation; the time of opening such scuttles in port and of closing and locking them before the vessel leaves port shall be entered in the official log; the construction of such scuttles shall be such as effectively to prevent any person opening them without the consent of the master.

(*c*) Scuttles fitted in the positions defined in clause (*a*) shall be provided with efficient metal shutters.

(2) In 'tween decks above the deck mentioned in paragraph (1) (*a*) of this article, opening side scuttles may be fitted except in spaces exclusively devoted to the carriage of cargo or coal.

(3) No side scuttles shall be fitted in any spaces which are exclusively devoted to the carriage of cargo or coal.

(4) All side scuttles which are not accessible during navigation shall be fitted with efficient metal covers, and both the glass and the cover shall be kept closed during navigation.

(5) No automatic ventilating scuttles shall be fitted in the ship's side below the margin line.

(6) All inlets and discharges in the side shall be arranged so as to prevent any accidental admission of water into the vessel.

(7) The number of scuppers, sanitary discharges, and other similar openings in the side shall be reduced to the minimum, either by making each discharge serve for as many as possible of the sanitary and other pipes or in any other satisfactory manner.

(8) Discharges led through the vessel's skin from spaces below the margin line shall be fitted with efficient and accessible means for preventing water from passing inwards. It is permissible to have either one valve, fitted with a means of working it at a distance, or two valves without such gear, one of these valves being always accessible. In either case, the accessibility of the valves or of the means of working shall be assured by their being situated above the deck referred to in paragraph (1) (*a*) of this article.

(9) In no case shall gangway, cargo, and coaling ports be fitted below the load waterline. None of these ports shall be fitted, even toward the ends of the vessel, in a space below the lowest 'tween-deck space in the amidship region of which it is permissible to fit such ports.

(10) Gangway, cargo, and coaling ports in the vessel's side below the margin line shall be effectively closed and made secure before the vessel leaves port, and kept closed during navigation.

(11) The inboard openings of ash shoots, rubbish shoots, etc., shall not be lower than the deck referred to in paragraph (1) (*a*) of this article. They may be permitted above this deck if fitted, to the satisfaction of the administration, with covers, which shall be watertight if below the margin line. Such covers shall be so arranged as to prevent their being clogged in any way, and shall be at least as easily and effectively closed as watertight doors and side scuttles.

CONSTRUCTION AND TESTS OF WATERTIGHT DOORS, SIDE SCUTTLES, ETC.

(1) The design and the materials used in the construction of watertight doors, side scuttles, gangway, coaling and cargo

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ports, valves, pipes, ash and rubbish shoots shall be to the satisfaction of the administration.

(2) Watertight doors shall be tested by a water pressure equal to that prescribed for the bulkhead where the doors are located. The test shall be made before the vessel is put in service, and either before or after the door is fitted.

CONSTRUCTION AND INITIAL TESTS OF WATERTIGHT DECKS, TRUNKS, ETC.

(1) Watertight decks, trunks, and ventilators shall be of the same strength as the watertight bulkhead at the place where they occur. The means used for making them watertight and the arrangements adopted for closing the openings in them shall be to the satisfaction of the administration. If watertight covers are used for closing these openings, they shall be fitted before the vessel leaves port, and kept closed during navigation.

(2) After completion a hose or reflooding test shall be applied to watertight decks and a hose test to watertight trunks. Watertight ventilators and trunks shall be carried at least up to the margin line.

(3) No change shall be made in the structure of watertight decks, trunks and ventilators after the survey without the permission of the administration.

PERIODICAL OPERATION AND INSPECTION OF WATERTIGHT DOORS, ETC.

In all vessels to which this convention applies, drills for the operating of watertight doors, side scuttles, valves, and closing mechanisms of scuppers, ash shoots and rubbish shoots, shall take place periodically during the voyage. A complete drill shall take place before leaving port, a second as soon as practicable after leaving port, and others thereafter at least once a week during the voyage, provided that all watertight power doors and hinged doors in main transverse bulkheads in use at sea shall be operated daily.

The watertight doors and all mechanisms and indicators connected therewith, and all valves the closing of which is necessary to make a compartment watertight, shall be periodically inspected at sea, at least once a week.

Hinged doors, portable plates, side scuttles, gangway, cargo and coaling ports, and other openings, which are required by the preceding rules to be kept closed during navigation, shall be closed before the vessel leaves port. The time of closing, and the time of opening (if permissible under these regulations), shall be recorded in the official log book.

A record of all drills and inspections required shall be entered in the official log book with an explicit record of any defects.

DOUBLE BOTTOMS

(1) In vessels 61 meters (equivalent to 200 feet) and under 76 meters (equivalent to 249 feet) in length, a double bottom shall be fitted at least from the machinery space to the forepeak bulkhead, or as near thereto as practicable.

(2) In vessels 76 meters (equivalent to 249 feet) and under 91.5 meters (equivalent to 300 feet) in length, a double bottom shall be fitted at least outside of the machinery space and shall extend to the fore and after peak bulkheads respectively, or as near thereto as practicable.

(3) In vessels 91.5 meters (equivalent to 300 feet) and over in length, a double bottom shall be fitted amidships and shall extend to the fore and after peak bulkheads respectively, or as near thereto as practicable.

(4) In vessels of over 91.5 meters (equivalent to 300 feet) in length, the inner bottom shall be continued out to the vessel's side in such manner as to protect the bilges.

(5) In vessels over 213 meters (equivalent to 699 feet) in length, the double bottom, for at least half the vessel's length amidships and forward to the forepeak bulkhead, shall extend up the vessel's sides to a height above the top of the keel not less than 10 percent of the vessel's molded breadth.

(6) Wells constructed in the double bottom in connection with the drainage arrangements shall not extend downwards from the inner bottom more than half the depth of the double bottom at that point. A well extending to the outer skin is, however, permitted at the after end of the shaft tunnels of screw vessels.

Every vessel defined shall be subjected at least to the following surveys: (*A*) A survey before the vessel is put in service; (*B*) Periodical surveys once each year; and (C) Additional surveys, as occasion arises.

RADIOTELEGRAPHY

All merchant vessels belonging to any of the contracting States, whether they are propelled by machinery or by sails, and whether they carry passengers or not, shall, when engaged on the voyages specified, be fitted with a radiotelegraph installation, if they have on board 50 or more persons in all.

Vessels on which the number of persons on board is exceptionally and temporarily increased up to or beyond 50 as the result of force majeure, or because the master is under the necessity of increasing the number of his crew to fill the places of those who are ill, or is obliged to carry shipwrecked or other persons, are exempted from the above obligation.

Moreover, the governments of each of the contracting States, if they consider that the route and the conditions of the voyage are such as to render a radiotelegraph installation unreasonable or unnecessary, may exempt from the above requirement the following vessels:

(1) Vessels which in the course of their voyage do not go more than 150 sea miles from the nearest coast.

(2) Vessels on which the number of persons on board is exceptionally or temporarily increased up to and beyond 50 by the carriage of cargo hands for a part of the voyage, provided that the said vessels are not going from one continent to another, and that, during that part of their voyage, they remain within the limits of latitude 30 degrees North and 30 degrees South.

(3) Sailing vessels of primitive build, such as dhows, junks, etc., if it is practically impossible to install a radiotelegraph apparatus.

Vessels which are required to be fitted with a radiotelegraph installation are divided, for the purpose of radiotelegraph service, into three classes:

First Class—Vessels having a continuous service.

There shall be placed in the first class vessels which are intended to carry 25 or more passengers—

(1) If they have an average speed in service of 15 knots or more;

(2) If they have an average speed in service of more than 13 knots, but only subject to the twofold condition that they have on board 200 persons or more (passengers and crew), and that, in the course of their voyage, they go a distance of more than 500 sea miles between any two consecutive ports. Nevertheless, these vessels may be placed in the second class on condition that they have a continuous watch.

Second Class—Vessels having a service of limited duration.

There shall be placed in the second class all vessels which are intended to carry 25 or more passengers, if they are not, for other reasons, placed in the first class.

Vessels placed in the second class must, during navigation, maintain a continuous watch for at least 7 hours a day, and a watch of 10 minutes at the beginning of every hour.

Third Class—Vessels which have no fixed periods of service.

All vessels which are placed neither in the first nor in the second class shall be placed in the third class.

The owner of a vessel placed in the second or in the third

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class has the right to require that, if the vessel complies with all the requirements for a superior class, a statement to the effect that it belongs to that superior class shall be inserted in the safety certificate.

Vessels which are required to be fitted with a radiotelegraph installation shall be required, by the governments of the countries to which they belong, to maintain a continuous watch during navigation as soon as the said governments consider that it will be of service for the purpose of safety of life at sea.

Meanwhile, the contracting parties undertake to require, from the date of the ratification of the present convention subject to the delays specified below, a continuous watch on the following vessels:

(1) Vessels whose average speed in service exceeds 13 knots, which have on board 200 persons or more, and which, in the course of their voyage, go a distance of more than 500 sea miles between two consecutive ports, when these vessels are placed in the second class.

(2) Vessels in the second class, for the whole of the time during which they are more than 500 sea miles from the nearest coast.

(3) Other vessels specified in article 31, when they are engaged in the trans-Atlantic trade, or when they are engaged in other trades if their route takes them more than 1,000 sea miles from the nearest coast.

Vessels connected with all kinds of fishing business, including whaling, which are required to be fitted with a radiotelegraph installation, shall not be required to maintain a continuous watch.

The radiotelegraph installations required above shall be capable of transmitting clearly perceptible signals from ship to ship over a range of at least 100 sea miles by day under normal conditions and circumstances.

Every vessel which is required to be fitted with a radiotelegraph installation shall, whatever be the class in which it is placed, be provided with an emergency installation, every part of which is placed in a position of the greatest possible safety to be determined by the government of the country to which the vessel belongs. In all cases the emergency installation must be placed, in its entirety, in the upper part of the vessel, as high as practically possible. The emergency installation includes an independent source of energy capable of being put into operation rapidly and of working for at least six hours with a minimum range of 80 sea miles for vessels in the first class and 50 sea miles for vessels in the two other classes.

If the normal installation, which in accordance with this article has a range of at least 100 sea miles, satisfies all the conditions prescribed above, an emergency installation is not required.

Every master of a vessel who receives a call for assistance from a vessel in distress is bound to proceed to the assistance of the persons in distress.

Every master of a vessel in distress has the right to requisition from among the vessels which answer his call for assistance the vessel or vessels which he considers best able to render him assistance, but he must exercise this right only after consultation, so far as may be possible, with the masters of those vessels. Such vessels are then bound to comply immediately with the requisition by proceeding with all speed to the assistance of the persons in distress.

The masters of the vessels which are required to render assistance are released from this obligation as soon as the master or masters requisitioned have made known that they will comply with the requisition, or as soon as the master of one of the vessels which has reached the scene of the casualty has made known to them that their assistance is no longer necessary.

If the master of a vessel is unable, or considers it unreasonable or unnecessary, in the special circumstances of the case, to go to the assistance of the vessel in distress, he must immediately inform the master of the vessel in distress accordingly. Moreover, he must enter in his log book the reasons justifying his action.

Life-saving Appliances and Fire Protection

At no moment of its voyage may a vessel have on board a total number of persons greater than that for whom accommodation is provided in the lifeboats and the pontoon life rafts on board.

The number and arrangement of the boats, and (where they are allowed) of the pontoon rafts, on a vessel depend upon the total number of persons which the vessel is intended to carry; provided that there shall not be required on any voyage a total capacity in boats, and (where they are allowed) pontoon rafts, greater than that necessary to accommodate all the persons on board.

Each boat must be of sufficient strength to enable it to be safely lowered into the water when loaded with its full complement of persons and equipment.

Any type of boat may be accepted as equivalent to a boat of one of the prescribed classes and any type of raft as equivalent to an approved pontoon raft, if the administrations concerned are satisfied by suitable trials that it is as effective as the standard types of the class in question, or as the approved type of pontoon raft, as the case may be.

The government of the contracting party which accepts a new type of boat or raft will communicate to the governments of the other contracting parties particulars of the trials made. It will also inform them of the class in which a new type of boat has been placed.

Embarkation of the Passengers in the Boats and Rafts

Suitable arrangements shall be made for embarking the passengers in the boats.

In vessels which carry rafts there shall be a number of rope ladders always available for use in embarking the persons on to the rafts.

The minimum number of sets of davits is fixed in relation to the length of the vessel; provided that a number of sets of davits greater than the number of boats necessary for the accommodation of all the persons on board may not be required.

Handling of the Boats and Rafts

All the boats and rafts must be stowed in such a way that they can be launched in the shortest possible time and that, even under unfavorable conditions of list and trim from the point of view of the handling of the boats and rafts, it may be possible to embark in them as large a number of persons as possible.

The arrangements must be such that it may be possible to launch on either side of the vessel as large a number of boats and rafts as possible.

The davits shall be of such strength that the boats can be lowered with their full complement of persons and equipment, the vessel being assumed to have a list of 15 degrees.

The davits must be fitted with a gear of sufficient power to insure that the boat can be turned out against the maximum list under which the lowering of the boats is possible on the vessel in question.

Any appliance may be accepted in lieu of davits or sets of davits if the administration concerned is satisfied, after proper trials, that the appliance in question is as effective as davits for placing the boats in the water.

The government of the contracting party which accepts a new type of appliance, shall communicate to the other contracting parties particulars of the appliance with details of the trials made.

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MEANS OF INGRESS AND EGRESS—EMERGENCY LIGHTING

(1) Proper arrangements shall be made for ingress to and egress from the different compartments, decks, etc.

(2) Provision shall be made for an electric or other system of lighting, sufficient for all requirements of safety, in the different parts of both new and existing vessels, and particularly upon the decks on which the lifeboats are stowed. On new vessels there must be a self-contained source capable of supplying, when necessary, this safety lighting system, and placed in the upper parts of the vessel, as high as practically possible.

(3) The exit from every compartment must always be lighted by an emergency lamp, which shall be kept locked, and which shall be independent of the ordinary lighting of the ship. These emergency lamps may be supplied from the independent installation referred to in the preceding paragraph, if an independent circuit is employed for this purpose and if this installation works concurrently with the ordinary lighting of the vessel.

CERTIFICATED LIFEBOAT MEN—MANNING OF THE BOATS.

There must be, for each boat or raft required, a minimum number of certificated lifeboat men. The minimum total number of certificated lifeboat men is determined by the provisions of the regulations.

The allocation of the certificated lifeboat men to each boat and raft remains within the discretion of the master, according to the circumstances.

By "certificated lifeboat man" is meant any member of the crew who holds a certificate of efficiency issued under the authority of the administration concerned, in accordance with the conditions laid down in the regulations.

FIRE PROTECTION

(1) The carriage, either as cargo or ballast, of goods which by reason of their nature, quantity, or mode of stowage, are, either singly or collectively, likely to endanger the lives of the passengers or the safety of the vessel, is forbidden.

This provision does not apply to the vessel's distress signals, nor to the carriage of naval or military stores for the public service of the State under authorized conditions.

(2) The government of each contracting party shall, from time to time by official notice, determine what goods are to be considered dangerous goods, and shall indicate the precautions which must be taken in the packing and stowage thereof.

(3) The following regulations indicate the arrangements to be made for the detection and extinction of fire:

1. A continuous patrol system shall be organized so that any outbreak of fire may be promptly detected.

2. Every vessel shall be provided with powerful pumps operated by steam or other means. On vessels of less than 4,000 tons there shall be two, and on larger vessels three of these pumps. The pumps shall be capable of delivering a sufficient quantity of water in two powerful jets simultaneously in any given part of the vessel, and shall be available for immediate use before the vessel leaves port.

3. The service pipes shall permit of two powerful jets of water being simultaneously directed on any given part of a deck occupied by passengers and crew, when the watertight and fireproof doors are closed. The service pipes and hoses shall be of ample size and made of suitable material. The branches of the pipes shall be so placed on each deck that the fire hose can be easily coupled to them.

4. Provision shall be made whereby both two powerful jets of water and a sufficient supply of steam may be conveyed to every space filled with cargo. Provision for the supply of steam need not be required in vessels of less than 1,000 tons.

5. A sufficient number of portable fluid fire extinguishers shall be provided, at least two being carried in each machinery space.

The governments of the contracting parties may accept other types of extinguishers provided that it is evident after trial that such extinguishers are as effective as the type referred to above. A government which accepts a new type of extinguisher shall send a description of the apparatus and particulars of the trial to the governments of the other contracting parties.

6. Two equipments, consisting of a smoke helmet and a safety lamp, shall be carried on board and kept in two different places.

7. All the fire-extinguishing appliances shall be thoroughly examined at least once each year by an inspector appointed by the government.

MUSTER ROLL AND DRILLS

Special duties for the event of an emergency shall be allotted to each member of the crew.

The muster list shows all these special duties, and indicates, in particular, the station to which each man must go, and the duties that he has to perform.

Before the vessel sails, the muster list shall be drawn up and exhibited, and the proper authority shall be satisfied that the muster list has been prepared for the vessel. It shall be posted in several parts of the vessel, and in particular in the crew's quarters.

Musters of the crew at their boat and fire stations, followed by boat and fire drills respectively, shall be held at least once a fortnight, either in port or at sea. An entry shall be made in the official log book of these drills, or of the reasons why they could not be held.

Different groups of boats shall be used in turn at successive boat drills. The drills and inspections shall be so arranged that the crew thoroughly understand and are practiced in the duties they have to perform, and that all the boats and pontoon rafts on the ship with the gear appertaining to them are always ready for immediate use.

GENERAL

This convention shall come into force on the 1st of July, 1915, and shall remain in force without any prescribed limit of time. Nevertheless, each contracting party may denounce the convention at any time after an interval of five years from the date on which the convention comes into force in that State.

This denunciation shall be notified through the diplomatic channel to the government of Great Britain, and by the latter to the governments of the other contracting parties. It shall take effect twelve months after the day on which the notification is received by the government of Great Britain.

A denunciation shall only affect the State which makes it, the convention remaining fully and completely operative as regards all the other States which have ratified it, or which have acceded thereto or which thereafter accede thereto.

This convention may be modified at subsequent conferences, of which the first shall be held, if necessary, in 1920. The place and time of these conferences shall be fixed by common consent by the governments of the contracting parties.

The governments may, through the diplomatic channel, introduce into this convention, by common consent and at any time, improvements which may be judged useful or necessary.

MATERIAL FOR BOSTON'S NEW DRY DOCK.—In deciding to build the immense new drydock, which forms part of Boston's port development, of reinforced concrete with wearing surfaces, or altars, of granite, photographs of reinforced concrete beams formed one of the determining features. These photographs showed that with proper mix and treatment the combined chemical and frost action of sea-water could be neutralized even when the material was subjected to alternate emersion under water and exposure to the air.

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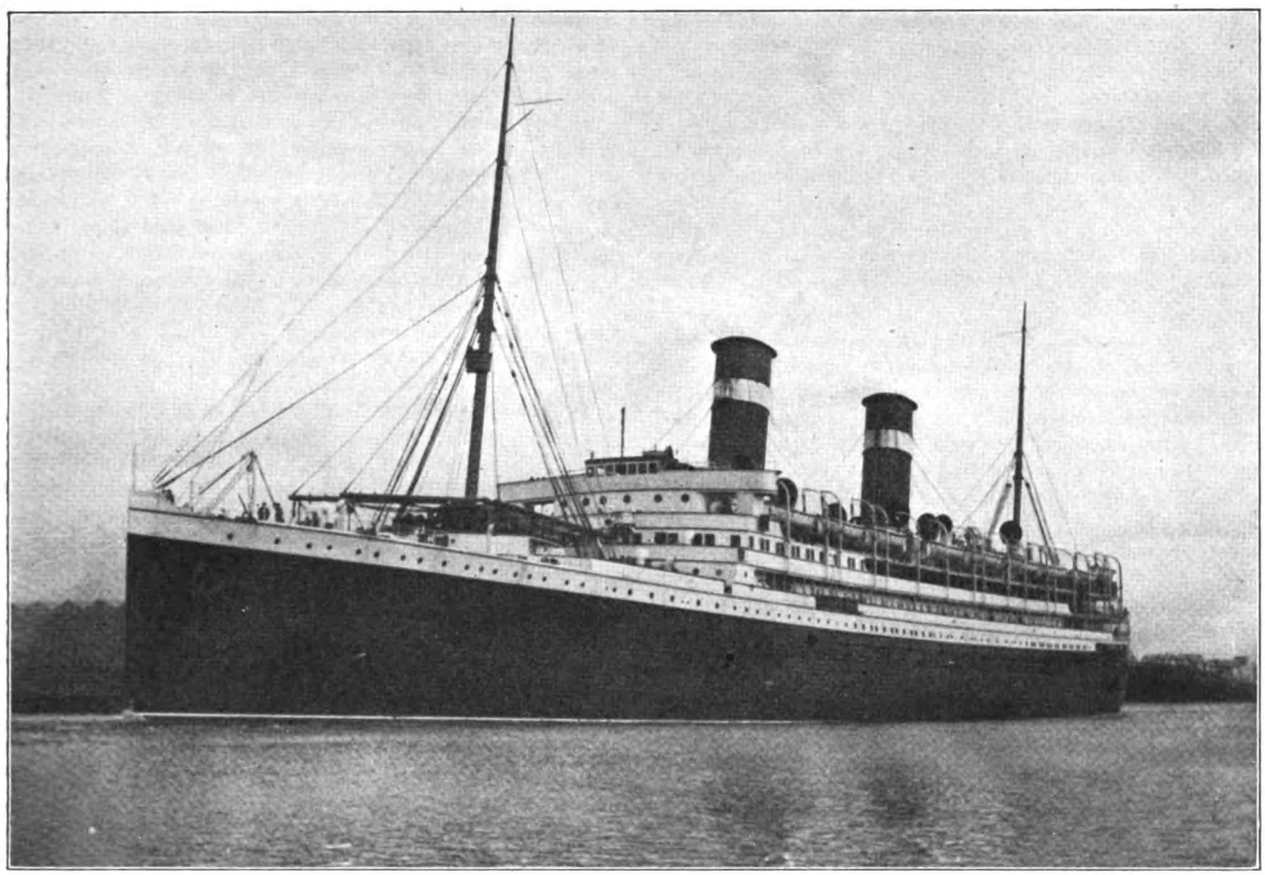
The New Allan Liner Calgarian

The Largest and Fastest Vessel Built for the Canadian Trade —Her Contract Speed Exceeded by 2¼ Knots on Trial

The quadruple-screw turbine mail and passenger steamer *Calgarian*, the largest and fastest vessel so far built for the Canadian trade, has been constructed for the Allan Line's main service between Liverpool, Quebec, and Montreal in summer, and Halifax and St. John in winter. The order for the hull and machinery was placed with the Fairfield Shipbuilding and Engineering Company, Limited, Govan, Glasgow. The vessel has been built to meet the requirements of the British Corporation Registry of Shipping, B. S.* class, also to Board of Trade and Canadian emigration rules.



New 18,000-Ton Passenger Liner *Calgarian* for the Canadian Trade

The massive appearance of the *Calgarian*, with cruiser stern, cruiser funnels, high pole masts and imposing superstructure, is seen in the photograph of the vessel given. She ran a series of speed trials on the Clyde March 16, when her best run was at a speed of 21.25 knots. The chief particulars of the ship are:

Length on waterline	590 feet 0 inch
Length between perpendiculars	570 feet 0 inch
Breadth, molded	70 feet 0 inch
Depth, molded to bridge deck	54 feet 0 inch
Mean draft	28 feet 6 inches
Gross tonnage	18,000 tons
Sea speed, designed	19 knots
Number of first class passengers	200
Number of second class passengers	500
Number of third class passengers	1,000
Number of officers and crew	500
Total number of passengers and crew	2,200

The propelling machinery consists of four turbines of the Parsons type, embodying the most recent improvements in design and construction, so as to ensure the maximum economy in fuel consumption, and resembling in this respect the machinery of the *Empress of Russia*, constructed recently at the same yard. The port outer shaft is driven by a high-pressure turbine, exhausting into an intermediate-pressure turbine driving the starboard outer shaft. The two inner shafts are each driven by a low-pressure turbine and they drive both ahead and astern. For maneuvering, when entering or leaving harbors independent high-pressure steam connections are provided on each low-pressure ahead turbine. There is also an independent high-pressure steam connection to the intermediate-pressure turbine. This, combined with an arrangement of valves, enables the high-pressure turbine to be cut out, or should the intermediate-pressure turbine be out of action, the high-pressure turbine can exhaust direct into either or both of the low-pressure turbines.

The whole of the main propelling and auxiliary machinery is situated in one watertight compartment. The turbine casings are of cast iron, and the drums, spindles, wheels and

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shafting are of forged steel. The four propellers are of the solid bronze type and each has four blades.

The condensing plant, which is placed in a separate watertight compartment immediately aft of the main engine room, consists of two main condensers of the Uniflux type, which, together with the two air pumps of the Dual type, are designed to maintain a vacuum of 28½ inches with the barometer at 30 inches and a sea temperature of 75 degrees Fahrenheit. There are four centrifugal circulating pumps of Messrs. Pauls' make. The air pumps discharge to filters of the gravitation type, through which the feed water gravitates to large float control tanks placed near the center line of the ship, and suctions are provided for the main and auxiliary feed pumps.

The feed heater is of the high-pressure type, two Weir evaporators have a total capacity of 150 tons per twenty-four hours, and for four distillers have each an output of 20 tons per twenty-four hours.

Steam is generated in six double-ended and four single-ended cylindrical boilers placed in two boiler rooms and worked under forced draft on Howden's system, having a total heating surface of 54,250 square feet and a total grate surface of 1,344 square feet, designed for a working pressure of 200 pounds per square inch. There are four electrically-driven forced-draft fans, each 6 feet 9 inches diameter, placed on the deck above the boilers. Ash hoists and ash ejectors are arranged in each boiler room. The coal bunkers are situated alongside the boiler rooms, and there are also coal bunkers across the ship at the forward and after ends of each boiler room.

The main electrical generating plant consists of three steam turbine-driven sets, each of 250 kilowatts capacity, while as a stand-by in case of complete breakdown of the main plant a small emergency turbo generator of 18 kilowatts is installed on the shelter deck, well above the waterline. The total number of lamps throughout the ship is about 3,000.

The sliding doors in the main watertight bulkheads below the upper deck are worked by the Stone-Lloyd hydraulic arrangement, and can be operated from the navigating bridge if necessary. The steering gear is of Brown's quadrant rack and pinion type, and placed on the *G* deck. There is also auxiliary and stand-by steering gear.

Stockless anchors are provided and are worked by two separate engines of Napier's make, each engine driving one cable holder and one warping capstan. There are also four steam warping capstans on the *D* deck at the stern, all of Napier's make. There are ten steam winches for working cargo. The ship is ventilated on the "Nuvacuumette" system, the fresh air being heated in winter and cooled in summer, ensuring a temperature of 70 degrees Fahrenheit in the ship.

The refrigerating plant is of the CO_2 type by the Liverpool Refrigeration Company; the refrigerated spaces occupy about 70,000 cubic feet. This is in addition to the ship's refrigerated provision rooms, which have a capacity of about 10,000 cubic feet.

The ship has eight decks, *A* the boat deck, *B* the promenade, *C* the bridge, *D* the shelter, *E* the upper, *F* the main, *G* the lower, and *H* the orlop deck. There are eleven watertight bulkheads, and it is claimed that the ship will remain afloat with any four adjacent compartments open to the sea. The double bottom extends all fore and aft, and is carried to the upper turn of the bilge.

In addition to the usual light and sound signals and an installation of wireless telegraphy, there is submarine signalling gear, and a bridge semaphore with Morse flashing lamp on a platform above the bridge. In the equipment of the vessel there is provided a 7-knot motor launch equipped with wireless telegraphy apparatus. In case of fog this launch will be sent ahead scouting, but will be secured to the steamer by about 400 yards of light steel wire so that her position will always be known. Lifeboats are provided for all, together with a life jacket for each person on board. There are 17 ordinary lifeboats and 28 Englehardt decked boats.

The passenger accommodation is arranged on the *A, B, C, D, E* and *F* decks. The first class berths have been designed in suites, special cabins, one-berth and two-berth staterooms. There are four sets of *en suite* cabins, consisting of two bedrooms, sitting room, bath and dressing room; each sitting room has two large couches, a writing table, concealed wash basin and suitable furniture. There are eight special cabins with bath rooms adjoining; the paneling and furniture of these rooms is in oak.

The public rooms include a dining room on the shelter deck, a library, lounge, card room and smoke room on the bridge deck, and a verandah café, upper smoke room and gymnasium on the boat deck. The style of decoration is Georgian throughout. The dining saloon extends the full width of the ship and has seating accommodations for about 190 persons, all at small tables. There is a large dome overhead in the center of the room, with a gallery all round and a balcony at one end for the orchestra. The lounge is an example of the more sumptuous decoration, fashionable in the time of George II., and it contains a fine carved mantlepiece and overmantle. There is a painted frieze of dancing children which runs all round the central part of the room. The smoke room is in French walnut, the mantle-piece, with reproductions of an old Vauxhall-glass mirror, is somewhat similar in character to that in the King's Room at Hampton Court Palace. The gymnasium is a lofty room, and is equipped with all the latest appliances for health-giving exercises.

Accommodation is provided for second class passengers in two and four-berth rooms, furnished like the first-class ordinary cabins, and having a large dining saloon, smoke room and lounge. The third class accommodation can be divided if required into two portions, so that either part can be placed in quarantine if necessary; a part of the third class space can be used for cargo purposes.

Externally the *Calgarian* resembles closely the Allan liner *Alsatian*, which was built by Messrs. Beardmore, Dalmuir, and described in our April issue of 1913. They are both of the cruiser stern design, but the rudders are not similar. The *Calgarian's* rudder is not supported outside the hull, while the *Alsatian's* rudder is hung on a single pintle in the stern post. The cruiser stern was severely tested during the turning trials of the *Calgarian*, and it earned unqualified approval because of its excellent qualities as regards freedom from vibration, ease of steering, and efficiency in propulsion. The ship turned a complete circle to port in 3 minutes 45 seconds, and a complete circle to starboard in 4 minutes 35 seconds.

The speed trials were run on the Clyde measured mile at Skelmorlie and consisted of seven double runs, beginning at a speed of 11 knots and working up gradually to a speed of 21.25 knots. On the last double run the mean speed was 20.63 knots, which is a knot and a-half more than that which will be required on regular service.

LLOYD'S QUARTERLY SHIPBUILDING RETURNS.—The returns compiled by Lloyd's Register of Shipping, which only take into account vessels the construction of which has actually begun, show that, excluding warships, there were 535 vessels of 1,890,856 gross tons under construction in the United Kingdom at the close of the quarter ended March 31, 1914. The tonnage now under construction is about 66,000 tons less than that which was in hand at the end of last quarter, and nearly 173,000 tons less than that building in March, 1913.

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The Watertight Subdivision of Ships and the Effect of Bilging

BY A. L. AYRE

The question of watertight subdivision in ships is one dealt with from two important points of view: First, technically, for the purposes of giving consideration to the intactness of the reserve buoyancy so as to minimize risk of foundering in the event of damage causing water to be admitted to the interior of the vessel, it being required that the quantity of water entering the vessel will be arrested at a certain point before the margin of reserve buoyancy has been consumed; secondly, commercially, for the purpose of giving cargo and passenger spaces within the boundaries of each compartment, of useful and economical length and capacity for the particular trade in which the vessel is to be engaged.

It will frequently be difficult to obtain ideal conditions in both cases, as it is quite apparent that a vessel so subdivided as to make her unsinkable after serious damage, may be one entirely uneconomical from a commercial point of view, owing to the fact that her cargo spaces may be too small and numerous, and consequently requiring separate hatchways and discharging appliances to each.

It is quite clear, however, that the benefit of any doubt should be given in favor of efficient watertight subdivision from the point of view of safety, and that in vessels designed to carry large numbers of passengers the question of safety should be the first to receive consideration. It will be interesting, first of all, to note how the buoyancy of a vessel is affected by bilging, also the condition of the vessel resulting therefrom. Fig. 1 shows a vessel floating in a normal condition at the waterline, *W. L.*, *G* being the center of gravity and *B* the corresponding center of buoyancy. The vessel being in equilibrium, these are vertically in line. Now, suppose the compartment *x, y, X, Y* to be bilged. The vessel will now lose the buoyancy of this compartment, which amount she must therefore recover by further immersion so as to regain the original displacement. The amount of mean sinkage is found by dividing the amount of lost buoyancy *x, y, X, Y* by the area of the intact waterplane, and for this purpose it is obvious that since none of the new displacement can be supplied between the bulkheads *X* and *Y*, this portion of waterplane area must therefore be left out of the calculation. The area of the intact waterplane will therefore be that contained between the points *W* and *x*, plus that contained between *y* and *L*.

Having obtained the amount of mean sinkage in this way it can be added on to the original mean draft, and the new mean draft is obtained, giving the waterline $W_1 L_1$, in Fig. 1. If the sinkage is only of small amount, the above result would be sufficiently accurate, but when it is large, the area of intact waterplane used for the final result should be a corrected one, taken at half depth of the layer of sinkage as previously found. This will allow for the vessel's change of form as immersion increases. It must be remembered, however, that should the damaged compartment contain cargo, an amount of buoyancy and area of waterplane, which will be contained in same, should be allowed for.

The vessel will not, of course, float at the new mean draft as found above, but will change trim according to the amended distribution of buoyancy resulting from the bilging of the compartment, because, although the new amount of buoyancy will be exactly equal to the former, the center of buoyancy will have altered so that the upward force will no longer be acting in the same vertical direction as hitherto. In Fig. 1 the new position of the center of buoyancy is shown by B_1, and this new position of the force of buoyancy acts with the force of gravity, as also shown, in such a manner as to cause the vessel to trim by the head and float at the waterline W_2, L_2. The question of change of trim will be dealt with later. A large amount of lost buoyancy causes a corresponding large amount of sinkage and probably also change of trim, and it is therefore seen that efficient watertight subdivision is essential so as to restrict the amount of buoyancy lost and the resultant detrimental consequences.

The main methods of obtaining watertight subdivision are:

1. Vertical transverse watertight bulkheads.
2. Vertical longitudinal watertight buikheads.
3. Watertight decks and flats.

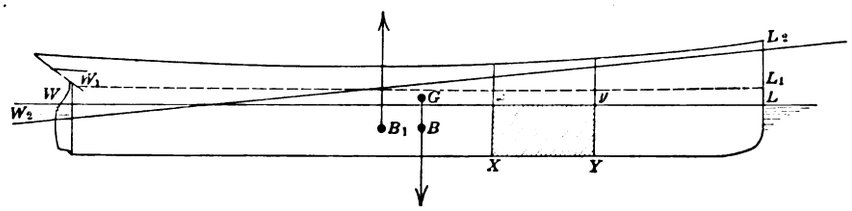

Fig. 1

4. Double bottoms subdivided transversely and longitudinally.

The first and fourth are, as a rule, the only means adopted in cargo vessels constructed for bulk carrying, the general desire being to have as large and spacious holds as possible, and when for any reason it may be convenient to have small holds for any particular trade in which the vessel may be engaged, it is generally more convenient to erect temporary wood bulkheads to divide off the quantities of cargo as may be required.

It would be very difficult to give any idea of the proportion of cargo steamers, having four transverse bulkheads and engines amidships, which, with the largest compartment bilged, could be expected to remain afloat, but it would certainly not be a large figure or proportion. In the case of small coasting steamers with engines aft, and generally without double bottom, we frequently find vessels of a length of about 150 to 160 feet having a hold about 90 to 100 feet long, and it is obvious that in the event of such vessels being bilged in the hold space, which covers about 55 to 60 percent of the total length, and situated in the fullest and most buoyant portion of the ship, they would be unable to remain afloat except when laden with cargoes which would enable them to float waterlogged. In many vessels of this class, however, it appears that nothing can be done to improve their fatal condition in the event of such bilging; for instance, many of them are purposely designed with long holds and hatches to enable them to carry long lengths of shafting or telegraph poles, etc., and another transverse bulkhead would, of course, entirely prevent them being suitable for the trade in which they are required to work. It is imperative that in the case of a vessel liable to be placed in such a plight that the most rigorous attention be paid to life-saving appliances, especially with a view to assuring the rapid and safe launching of boats, etc.

Frequently the conditions of ordinary ocean-going cargo steamers are very little better; for instance, a vessel of the smaller class, say 280 feet long, with engines amidships and

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having four transverse watertight bulkheads, may have a forward hold space equal in length to 35 or 40 percent of the total length of ship. Should such a vessel be bilged in the laden condition with a non-floatable cargo, it is obvious that she also must be in a serious predicament.

The following example of such a vessel will show this: The dimensions of the vessel are 280 feet between perpendiculars by 42 feet molded beam by 21 feet 6 inches molded depth and 18 feet 6 inches molded draft. The block coefficient is .8, giving a displacement, including shell, of 5,000 tons. Fig. 2 represents the vessel.

If the forward hold is bilged the amount of lost displacement shown by the shaded portion in the sketch is:

$\frac{120 \times 42 \times 18.5}{35} \times .86$ (coefficient of fineness of this portion) $= 2{,}291$ tons.

It must be remembered, however, that this is not the correct amount of buoyancy, because within the shaded portion there is situated some cargo and structure which itself provides some buoyancy, owing to the fact that it prevents the water from occupying the whole of the space. In the case of a compartment filled with coal, water can find access to about 40 percent of the space. To allow for cargoes of greater density than coal a reasonable figure to take is 50 percent. In the above case the amount of hold space in way of the displacement represented by the shaded portion is 75,000 cubic feet, and in view of the above we can take half of this as being buoyant when bilged.

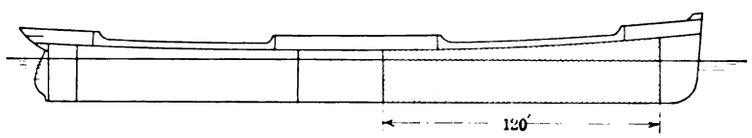

Fig. 2

$\frac{75{,}000}{2} \times \frac{1}{35} = 1{,}071$ tons, then $2{,}291 - 1{,}071 = 1{,}220$ tons, net loss of buoyancy.

The area of the intact waterplane divided by 35 will give the "tons per foot," and the amount of lost buoyancy being then divided by this will give the amount of sinkage in feet.

Total area of waterplane $= 280 \times 42 \times .9 = 10{,}584$ square feet. A deduction must now be made for the waterplane area in way of the bilged compartment, and allowance must here again be made in respect of the cargo lying in way of same, which, since it provides buoyancy, will also provide waterplane area. The amount of area provided will be in the same proportion as previously taken for buoyancy in way of cargo, *i. e.*, 50 percent. The amount of waterplane area which is lost in way of the bilged compartment is therefore $120 \times 42 \times .95$ (coefficient) $\times$ 50 percent $= 2{,}394$ square feet. The net area of intact waterplane now becomes $10{,}584 - 2{,}394 = 8{,}190$ square feet.

The "tons per foot" displacement at the load waterline in the damaged condition will be $8{,}190 \div 35 = 234$ tons.

The extent of the *mean* sinkage owing to the loss of buoyancy in the damaged compartment is found by 1,220 tons, net loss of buoyancy $\div$ 234 $= 5.2$ feet.

Original load draft $= 18.5$ feet molded
$+$ mean sinkage $= 5.2$ feet
23.7 feet $=$ mean draft in damaged condition.

Since the molded depth of the vessel is only 21 feet 6 inches, it is seen that the mean draft in the damaged condition is greater than this by 2.2 feet; but it must be remembered that in obtaining this new mean draft we have not allowed for the change in the amount of "tons per foot' which occurs after the deck becomes immersed, *i. e.*, when the buoyancy remaining is that contained in the sheer and in the poop, bridge and forecastle.

In the above example about 1,350 tons of displacement are contained in the hull proper above the load line and up to the deck including the sheer, this, together with the load displacement of 5,000 tons, making in all 6,350 tons. The 1,350 tons of reserve is equal to about 21 percent of the total amount, while for a vessel of this molded depth the British Board of Trade tables require 27.3 percent. The difference in the example is, of course, taken as being made up in the erections, and, as a matter of fact, in the case taken it is more than made up if the erections are taken at the normal height of 7 feet, and as being perfectly watertight, as the percentage of reserve buoyancy then actually works out at a little over 31. The additional buoyancy contained in the erections is about 918 tons on this basis.

In the actual circumstances, however, 1,350 tons of reserve buoyancy does not exist in the hull proper, as there is a total of about 680 tons contained within the damaged compartment, and if only 50 percent of this, *i. e.*, the buoyant value of the cargo, is taken, the net amount of reserve buoyancy will then become $1{,}350 - 340 = 1{,}010$ tons.

The net loss of buoyancy being 1,220 tons, it is seen that even with the whole of the hull immersed there is still a further amount of 210 tons required to be made up by the erections.

Seeing that these erections are rarely intact as regards watertightness, it is at once seen that little likelihood remains of there being sufficient reserve of buoyancy to balance the amount which has been lost in the bilged compartment. The large change of trim and the loss of transverse stability, both of which will also occur, will further jeopardize the vessel, and it is quite clear that she cannot by any means be expected to remain afloat, and it is also seen that a vessel with the bulk of her reserve buoyancy contained in the hull proper, instead of depending largely upon the amount being made up in the erections, will be a much safer vessel, even comparing with a case in which the erections are perfectly watertight.

The number of bulkheads are generally determined by arranging them in such positions that no compartment will exceed a certain maximum length, or to determine the number in accordance with the length of ship. While in purely cargo vessels such methods may give, as it were, a compromise in the consideration of watertight subdivision from the safety and commercial points of view, they should not be *tolerated* in passenger ships as a means of determining their subdivision.

In the latter cases the spacing of bulkheads and other means of watertight subdivision should be such that will provide safety together with a margin when a certain percentage of length is flooded, this percentage of length being reasonably large, and, of course, having further subdivision within itself so as to minimize the effect of the bilging as far as possible. As seen in the foregoing example, and in the case of single-decked vessels, or other vessels without a watertight lower deck, it could not, of course, be expected that in ordinary cases a laden vessel with 40 percent or more of her length exposed to the sea, could remain in a floating condition, and it appears, therefore, that the aid of the watertight lower deck, while not limiting the extent of flooding longitudinally, will, as a rule, do so vertically, and thereby reduce the amount of flooding and enabling the buoyancy lying above same to be preserved intact. A watertight lower deck in passenger vessels will therefore be a substantial advantage and a most valuable asset on the side of safety.

(To be continued)

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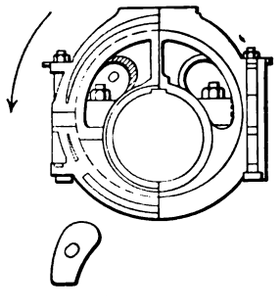
Letters from Practical Marine Engineers

Incidents Relating to the Design, Care and Handling of Marine Engines, Boilers and Auxiliaries; Breakdowns at Sea and Repairs

Repairs to a Broken Eccentric at Sea

The go-astern eccentric is keyed to the shaft and its position is fixed and permanent. On the other hand, the go-ahead eccentric is not keyed to the shaft, but is driven by the go-astern eccentric, a bolt passing through both eccentrics for this purpose. This bolt is a driving fit in the go-astern eccentric, but passed freely through an opening in the go-ahead eccentric. This space around the bolt allows for adjustment to increase or decrease the angular advances of the eccentric at will.

After the go-ahead eccentric is adjusted to the desired position the place around the bolt is filled in with hard wood



Sketch of Repairs

blocks and the nut is put on and set up tight. In one instance these blocks of wood became loose and crumbled, allowing the bolt to swing back and forth in the slotter opening, breaking the arm or the rib of the eccentric.

Repairs were made in this case by fitting a strap of half-inch boiler plate between the rib or arm of the eccentric and the part where the two halves of the eccentric are bolted together, securing the strap in place by the nut of the driving bolt. The strain of driving the eccentric then came on the metal where the two halves of the eccentric were joined.

Philadelphia, Pa. J. E. C.

Corrosion of Boilers through the Use of Sulphurous Oils

On several occasions the excessive corrosion of boilers using oil-fuel having a high sulphur content has been noted in current engineering literature. Since many seen to be quite unable to account for this corrosion, I believe that this letter setting forth what I believe to be the correct theory will be of interest. Needless to say, this thory is not entirely original, so many may be acquainted with some part of it.

The following extract is quoted from Technical Paper No. 26. Bureau of Mines: "The corrosive action of sulphur acids on fireboxes and boiler flues, and on the cylinders of internal combustion engines, is not thoroughly understood. The effects of such action are sufficiently evidenced by the pittings, scales, and roughened spots on metal surfaces that have been exposed to the products of combustion of fuels rich in sulphur. Therefore an exact knowledge of the sulphur content of fuels is desirable."

Reference to Bulletin No. 19, of the same Bureau, will give one that exact knowledge of many of the California oils. For the purposes of discussion these are as suitable as any. On the average, the West coast oils are quite rich in sulphur. An examination of the analyses given in the above-mentioned bulletin (oils of the San Joaquin Valley of California) shows a sulphur content varying from .12 percent to 1.32 percent. The average sulphur content of the oils of this vicinity seems to be about .8 percent. This may seem small to many, but if one will remember that for every ton of fuel burned there is also burned, approximately, 18 pounds of sulphur, he will see that the weight of sulphur burned in a large furnace, in the course of a few days or weeks under steam, will mount to rather astonishing figures.

The sulphur existing in the liquid fuel is burned with the oil as it passes into the furnace. When sulphur burns, in air or in oxygen, it unites with oxygen, forming a large amount of sulphurous oxide (SO_2) and a small amount of sulphuric oxide (SO_3). Both of these gases are colorless and therefore cannot be detected by observation. Each of them can have a corrosive effect on such metal as it may come in contact with, while passing through the boiler.

At high temperatures, in the absence of air, iron dissolves in SO_2, forming ferrous sulphite ($FeSO_3$). It is therefore quite probable that this reaction may go on to a slight extent at furnace temperatures, and since large quantities of SO_2 are present, this may cause marked effects in the course of time.

Under favorable conditions SO_3 unites with iron to form ferrous oxide (FeO). It is, however, quite improbable that this reaction takes place to any extent, on account of the great dilution of the SO_3 in the products of combustion.

As the effect of a slight covering of soot on the tubes would greatly reduce any reactions from the above causes, it is with the further products formed by the SO_3 that we are principally interested, since it is from this source that the chief cause of trouble originates.

When formed, the SO_3 is in the gaseous state. At 46 degrees C. (114.8 degrees F.) it condenses to a liquid, and at 14.8 degrees C. (58.6 degrees F.) it solidifies into a white wax-like solid. As a matter of course, the temperature of the products of combustion will not get as low as 46 degrees C. while the boiler is under steam, therefore the SO_3 will remain as a gas during this time.

It seems to be an established fact that gas is occluded, that is, absorbed and retained, by firebrick, porous lagging, etc. This is not at all surprising, since many substances possess truly remarkable powers of occluding large quantities of gas. In the case at hand, I believe that the porous substances in the boiler, that is, firebricks, baffling, and lagging, absorb and retain large quantities of SO_3. Some small part of this gas may be given off as the boiler cools, but, since occluded gases are not easily given off, the greater part of the gas will remain in the absorbing medium.

As soon as the temperature falls below 46 degrees C. the contained SO_3 will become a liquid, and, having a specific gravity of approximately 1.94, will tend to flow toward the lower part of the absorbent. If the temperature continues to fall, the SO_3 will solidify at 14.8 degrees C.

In either the liquid or the gaseous form, the SO_3 unites readily with water to form sulphuric acid, according to the following equation:

$$SO_3 + H_2O = H_2SO_4.$$

Since, ordinarily, water vapor can always be found in the combustion spaces of a dead boiler, an ample supply of water is easily forthcoming by the condensation of the water vapor on the metallic surfaces. The product is then a dilute sulphuric acid.

Dilute sulphuric acid is a most active reagent with iron of any kind, uniting with it to form ferrous sulphate, as follows:

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$$H_2SO_4 + Fe = FeSO_4 + H_2$$

The ferrous sulphate thus formed is in the form of green crystals and should be easily detected upon a critical examination of a corroded part. The compound thus formed is composed of .3675 parts of pure iron to .6325 parts of other elements; that is, it is more than one-third pure iron. From this it is easily seen that a small quantity of the sulphate may indicate serious pitting if the action is localized. In all probability, action from this cause will be extremely local, usually taking place at such places as the absorbent materials are in contact with metallic surfaces, more especially when this contact is near the bottom of the absorbing material.

If it is desired to test either scales scraped from a metallic surface or pieces of firebrick to determine whether or not sulphuric acid or sulphates are present, scrapings or small pieces of the bricks should be put in a small quantity of distilled water and allowed to remain there a few minutes. After the soluble matter is dissolved, the liquid should be separated from the solid matter either by filtration or by decantation; that is, by allowing the solids to settle and then carefully pouring off the liquid. A few drops of barium chloride ($BaCl_2$) should now be added. If either sulphuric acid or ferrous sulphate is present in the solution, a dense white precipitate of barium sulphate will immediately appear. This is a very sensitive, quick, and satisfactory reaction.

As a means of preventing further trouble from this source, the following suggestions are made, both of which are quite apparent. First, keep the boiler dry when not in use, since the acid cannot be formed without water. Just how completely the formation of acid can be prevented by careful attempts to keep out moisture remains to be demonstrated by experience. Second, sprinkle ordinary lime, either quick or slaked, upon the troubled areas, after the boiler has cooled. Such acid as may come into contact with the lime will be neutralized, without damage to the iron. W. W. B.

Serious Accident Due to Shortness of Water

To all marine engineers the boilers are the most important section of their department, and it cannot be too deeply impressed on the minds of those in charge the serious results that may be incurred by the neglect of the water gage glass, and that a large proportion of boiler troubles are caused by shortness of water. The writer's experience showed that in one case, anyhow, a serious accident would have been averted had proper precautions been taken in the first instance.

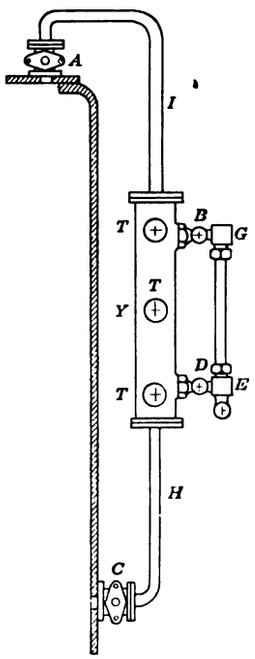
We were about to leave Cardiff on a voyage down the Mediterranean and Black Sea ports, when our fourth engineer (who, by the way, was continually growling about the misery of a marine engineer's life) failed to show up on the morning of sailing, so we either had to go to sea short-handed or else seek a substitute. The latter course was favored by the chief engineer for obvious reasons, so our own donkeyman, who was a trustworthy man, but did not hold any certificate or license, was engaged.

After a couple of four-hour watches, in which the chief helped the donkeyman out, he was left to go below alone, in sole charge of the engines and boilers. I myself had to relieve him at midnight and all was well.

One night several days out, however, about 11 P. M., I was aroused from my meagre slumbers by a terrific noise as of escaping steam somewhere in vicinity of the fire room, which caused me to dress hastily and rush from my berth. At the top of the fiddley grating I was met by an oiler who acquainted me with a hurried explanation of the explosion. He said the starboard boiler's high furnace had burst.

Proceeding down the ladder to the stokehold, I saw the chief, second, and some firemen attempting to close the starboard boiler stop valve and auxiliaries. The heat was intense on the tops of the boilers, and I remember the top button of my boiler suit burned my chin when it came in contact with it, raising a blister. It was useless to attempt putting water into the boiler, as the combustion chamber top had collapsed and was down about 4 inches, the nuts stripped off eight of the staybolts supporting it, and six of them were drawn through the plate, leaving six 1½-inch diameter holes through which the steam escaped with great violence, the pressure carried being 180 pounds per square inch.

There was no doubt that the boiler became short of water, the combustion chamber crown plate was overheated and was drawn over the ends of some of the supporting stays. Our erstwhile fourth engineer swore there was plenty of water



Arrangement of Water Column

in the starboard boiler, and upon examination this proved to be incorrect, although the glass was full. The trouble lay in the top double shut-off cock or valve becoming choked with deposit, allowing the gage glass to fill up.

The test cocks on marine boilers in the writer's experience have usually been for ornamentation, as they are packed tight to prevent leakage, and difficult to open, so consequently never touched. Now, to help junior engineers who are not fully conversant with the methods of testing water gages, I propose to give a short, reliable way to ascertain if all pipes, cocks, valves, etc., on gage columns, etc., are clear.

To blow glass through shut *B* and *D* alternately and keep *E* open, but to blow through the water gage it is necessary, after blowing through the glass, to shut *A* and *C* alternately, at the same time keeping *B*, *D* and *E* open long enough to completely discharge the contents of the gage and its connections. Suppose *A* is choked, assuming *B*, *D* and *C* are clear. The steam in the column and in the pipe *I* becomes condensed, and the water flowing through *C* to take its place rises in column *Y* and in the glass to a level higher than the water in the boiler. This means a false level. It we now open *E* and water is blown out, then on *E* being closed again the water will rise higher again than before misleading the engineer further.

When *B*, *D* and *A* are clear and *C* choked, then any water that may be in the glass is trapped and no longer rises and falls with the water in the boiler, or with the motion of the ship. It will, however, rise in the glass, owing to the con-

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densation of the steam in the upper part of the gage until *E* is opened, when all the water in the glass is blown out. On closing *E* the glass shows no water, although the water in the boiler may be at the working level.

When the test cocks *T*, *T*, and *T* are attached to column *Y*, they cease to be reliable when either double shut-offs *A* or *C*, or the pipe in connection, is choked or partially so.

Woodbridge, N. J. F. W. CHRISTIANSEN.

Securing a Loose Piston

Quite recently, while running at about 180 revolutions per minute, approximately fifteen knots, a very loud, sharp knock was heard in the vicinity of the starboard after low-pressure cylinder. This knock was so distinctly metallic that it was known at once that something had come adrift. The engine was immediately stopped, and all crank and crosshead bolts and nuts examined. All were found tight. However, when the engine was slowly started again, the same knock continued.

Both engines were then stopped. The jacking gear was thrown in on the starboard engine, after which the port engine was slowly worked up to speed. The starboard engine was jacked over until the low-pressure piston was at the top of its stroke. As a matter of precaution, blocking was put in, between the lower cylinder heads and the crossheads, wherever the dragging propeller tended to make the crossheads rise. By securely wedging these short shores, a small part of the strain was taken off the jacking gear.

As soon as the cylinder was sufficiently cool to allow work to proceed, the after low-pressure cylinder head was removed. Investigation showed that the piston nut had backed off about a quarter of an inch, and that everything else was apparently in good condition.

An attempt was made to set up on the piston nut, but this was not successful. It was soon found that the threads had been so jammed, by the pounding of the piston, that the limit of travel of the nut was only about one-half of a turn. This gave about a quarter of an inch vertical motion.

Since the vessel was only about seventy-five miles from the end of her run, it was decided to try quick repairs, as an experiment, without any thought of having them hold longer than necessary to finish the trip.

For emergency repairs, three horseshoes were cut out of heavy sheet lead. After backing the nut off as far as possible, the lead horseshoes were slipped into place between the top of the piston and the nut. In each case the horseshoe was somewhat closed by bending in the long points, thus making almost a complete ring. These three rings made a snug fit, practically filling the entire opening. The nut was then set up, using a heavy maul, until it was quite firm. The cylinder head was replaced, and, after removing the jacking gear, shores, etc., the engine was started at moderate speed.

By keeping the auxiliary exhaust out of the low-pressure cylinder, so that the temperature remained quite low, it was hoped that these lead rings would stand the compression to which they were subjected, until arrival in port. The engine did run it, but by the time of arrival in port it was pounding quite badly.

After arrival in port, the cylinder was again opened and the temporary lead rings removed. Needless to say, some small bits of lead were adrift in the cylinder, but they were not large enough to cause any trouble.

Two horseshoes, having long points, were then cut out of sheet steel. They were only roughly finished, but were made to be a good fit for the upper end of the piston rod. These horseshoes were slipped into place, after the nut had been backed off, and the ends were bent down so as to catch on the side of the boss on the piston. This preliminary bending kept them from working out when the nut was set up. The nut was then set up with great force; in fact, to what we believed to be about its limit of travel without rupture. After this, the points on the horseshoes were well hammered down and the cylinder was closed as usual.

This engine was put into service the next morning, after which it ran, without trouble, until opportunity offered for making permanent repairs. It was operated at speeds as high as 220 revolutions per minute without showing any signs of working loose around the repairs described.

Although, in this case, sheet lead gave fair results for a very temporary job, it certainly is not to be recommended for any such work, since it is far too easily flattened by the pressure necessary to get much work out of that cylinder. As soon as the job gets sufficiently loose, so that the lead receives a blow each stroke, the repairs very quickly go to pieces. It should also be remembered that this was the low-pressure cylinder of a triple-expansion engine, and therefore working at a comparatively low temperature. In a cylinder in which the working heat is high, I do not believe that such a repair job would give sufficient satisfaction to justify its installation. This kind of a job was used here only because the run would have been finished before any other repairs could have been made, and because it was believed that it would hold sufficiently well to allow the engine to be used while working around the docks.

When permanent repairs were made it was found necessary to split the nut in two places before it could be removed. The threads on the piston rod were found practically uninjured, while those in the nut were badly jammed and partially stripped. A new nut was made and installed, thus completing repairs. W. W. B.

Friction

Friction is defined as the resistance to sliding, which is experienced when one body slides over another. A well-known text book on physics says "Friction is no doubt due to interlocking of the projections on one surface with those on the other surface."

There is a rather widespread conviction among engineers that the above statement accounts for friction, but a moment's consideration will show that friction cannot be entirely due to the above cause. The second law of friction states that "Friction is independent of the area of the surfaces in contact, for any given normal pressure, i. e., it is independent of the intensity of the normal pressure." Now, if friction were due to the interlocking of the projections, it will be seen that friction will become dependent on the area of the bearing surfaces in contact, which is in direct opposition to the second law.

Again, with interlocking of the projections, when one surface slides over another we would get abrasion of the surfaces taking place and the laws of friction would not hold. If the two bodies were perfectly smooth, there would still be a certain amount of resistance to sliding, hence it would seem that friction is really due to molecular cohesion of the particles of each surface.

The reasons in support of this theory may be indicated as follows: Matter is composed of molecules which attract one another with tremendous forces. These forces, however, are only sufficient to hold each particle of the body together through an infinitely small distance. When a bar of iron, say, is broken in tenison and the two broken ends fitted together again, the application of sufficient force to bring the two portions close enough together would cause the pieces to reunite to form a solid bar. Of course it is impossible from practical considerations to join up a broken bar by this means, but a similar case is found in the forming of powdered graphite into a solid block by the application of pressure.

These molecular forces, however, will be partly felt at the small distances attainable in practice, hence the theory that friction is due to molecular cohesion of the particles would seem to have some weight. "ISON."

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Marine Articles in the Engineering Press

Commercial Importance of the Panama Canal.—By Emory R. Johnson, Ph.D., Sc.D. In this paper the author shows the commercial importance of the Panama Canal, or the assistance it will render industry and the trade of the people of the United States and other countries, first by showing what effects the new route will have upon the length and time of ocean voyages, then by stating conservatively the volume of shipping, foreign and American, international and coastwise, that may be expected to use the canal. Estimates are given to show the influence which the Panama Canal will have upon the freight rates by rail between the two seaboards of the United States and by ocean carriers engaged in the American intercoastal and foreign commerce. Other points covered are fuel costs, in which the author points out how much the ocean carriers will save in fuel costs by using the Panama route, and how the cheaper fuel costs by way of the Isthmus will assist the Panama Canal in competing with alternative routes via the Straits of Magellan, the Cape of Good Hope and the Suez Canal. Finally, the relation of tolls to the traffic and revenues of the canal is analyzed. 9,100 words.—*United States Naval Institute Proceedings.* December.

The Story of the Orient Line.—Unlike most of the large British steamship companies, the Orient Line came into existence with a few chartered steamships with which a new line was established in 1877 between England and the Australian colonies over the long sea route around the Cape of Good Hope. This line was established without government assistance, and gradually increased its activities with the addition of new and up-to-date vessels, and with the introduction, from time to time, of new innovations, such, for instance, as the provision of refrigerated space in their vessels. The latter innovation was made in 1880, and from this beginning there sprang the huge business in meat, perishable produce and fruit which now exists between the commonwealth and the mother country. At about this time, the Cape route was deserted for that through the Suez Canal and Red Sea to Colombo. The establishment of regular schedules, with speedy up-to-date vessels, soon acquired for the company a share in the mail contracts, of which the company continually took advantage, as, for instance, when in 1908 the Post-Office insisted, among other things, upon the provision by the contractors of a new fleet of 12,000-ton vessels, a condition which the Orient Company readily met. The Orient Line has never engaged its ships in any other regular service than that to Australia. The main source of its activity, outside the mail service, has been the tourist business, to which this company was one of the first to give serious attention. 9 illustrations. 2,800 words.—*The Marine Engineer and Naval Architect,* January.

Considering Trim of Ship on Design of Lines.—It is pointed out that modern demands for certain predetermined qualities of the ship are much greater now than formerly, and that frequently trim, say for light or ballasted conditions, has to be estimated at the time of the design of the lines. The starting point is said to be the center of gravity for different drafts and conditions, thereby also determining the position of the center of buoyancy fore and aft. From trimmed conditions it is possible to advance to the center of buoyancy for level keel conditions by estimating the longitudinal metacentric radius, and with the proper trim angle the horizontal shift of the center of buoyancy it is assumed that the vertical displacement scale, the curve of waterline areas and therewith the vertical position of the centers of buoyancy are given. Finding as above outlined the horizontal position of the centers of buoyancy and entering them to scale in their actual positions on the curve sheet, then the actual horizontal positions of the water planes are shown to be the intersections of the waterlines with tangents through the respective center of buoyancy to the whole curve of actual centers of buoyancy. The area of the water planes being given, and its individual center of gravity being found, no difficulty is experienced in laying down the lines according to the requirements. 4 illustrations. 1,140 words.—*Schiffbau,* January 28.

German River Shipping in 1913.—The greater regularity of the volume of the German river shipping is explained, although statistics depend a little uncertainly upon a census taken only every five years, except on the Rhine, where a registered society issues a fairly complete list of river vessels. The information on river and canal tonnage was somewhat inaccurate up to 1912, but in 1913 the "Central Association of German River Shipping" issued quite an accurate register. It states that 1,196 river vessels of 266,386 tons carrying capacity were built, the total accounting for more ships but less tonnage than in 1912. Of this Holland shipyards furnished seventy-two vessels of 43,634 tons for German account, a competition very distasteful to many interests of the region of the lower Rhine. To offset this, financially, it is recorded, with satisfaction, that some of the German Rhine shipyards built a considerable number of high-grade vessels for foreign account of higher value than the addition from Holland yards. The Rhine district shows a reduction of tonnage against 1912, but an increase in steam and motorship horsepower. In the Ems district there is a reduction of almost one-half recorded. In the Weser and Oder districts the barges built were of practically the same tonnage, while steamer and motorship building was materially reduced. In the Elbe district tonnage decreased but the horsepower of steamers increased. In the Eastern districts the tonnage remained practically constant. 1 table. 1,320 words.—*Schiffbau,* January 28.

Tariff, Percentage or Trust in Shipbuilding.—A full discussion of the economic conditions is given, under which the leading incorporated German shipyards are so severely handicapped that they pay either no dividend at all or at least only an unsatisfactory one. Only one large private yard is on a satisfactory financial footing. A number of remedies are discussed, taking up in turn the questions whether help could be expected from a brighter future, more extended credit from the banks or special orders and assistance from the government. But only three measures for the betterment of the financial conditions are estimated as worthy of serious consideration, namely, a tariff on all ships built in foreign countries, a percentage agreement on builders' profits and a trust for regulating prices. The first possibility, a protective tariff on ships, whether directly levied or in form of rebates on harbor dues or railroad freights to home vessels, or increased taxes on foreign flags or tonnage is, however, considered of doubtful value, as seagoing shipping must compete in the international market with all nations and a tariff would be a severe handicap. The second possibility deals with agreements between builders and owners of ships upon a fixed percentage of builders' profits on and above the actual cost of the vessel, pointing out, however, that this would reduce the shipyard more or less to a subordinate employee of the owner and be seriously dulling the ambition and incentive of the shipyard to produce work at the minimum cost. A number of such firms are mentioned, among them being in the United States the Standard Oil Company and the Great Northern Railroad, who are stated to own shipyards in Camden and

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New London. The third possibility is that of a trust of all, or at least all, large shipyards, which is intimated would prove a very satisfactory solution of the question of insufficient earnings. It is assumed that the management of the syndicated shipyards would not increase the price on merchant ships that have to compete in the international freight market, but on the giant passenger liners and on warships. It is suggested to increase prices, because, in the first place, a wealthy patronage could bear increased prices, and, in the second place, the government and the entire taxpaying community could carry the burden without feeling it. 3,500 words.—*Schiffbau,* February 11.

Oil Tanker Jupiter.—The tanker *Jupiter* was launched in January at the Howaldt Works near Kiel for the German-American Petroleum Company in Hamburg. She is not only the largest single-screw steamer ever built, but also the largest tanker, exceeding by about 2 feet beam similar English-built tankers. She is 525 feet long by 68 feet 6 inches molded beam, by 41 feet 9 inches depth by 26 feet 6 inches draft, and of about 15,000 tons carrying capacity. She is built as a shelter-deck vessel under the highest class of English Lloyd's, after the longitudinal system of framing with the propelling machinery in the stern. There are seventeen transverse bulkheads, and the hull is divided into twenty oil compartments, bound by cofferdams at the ends with the pump room in the middle. An expansion tank, 41 feet wide with summer tanks at the sides, extends over the entire 295 feet 3 inches length of the tanks. To balance the engine weight a package freight cargo hold and a ballast or oil tank are provided forward. The pumping plant consists of two horizontal duplex steam pumps of 430 tons per hour capacity and a large turbo-fan for removing the gases. The engine plant consists of one four-cylinder engine of 3,500 indicated horsepower at 72 revolutions per minute, giving the ship a speed of 10½ knots; the cylinders are 27.2, 40, 57.1 and 81.3 inches diameter by 40 inches stroke. One air pump, two feed, two bilge, one sanitary and two evaporator pumps are driven from the crossheads. There are three single-ended boilers, 15 feet 5 inches diameter by 11 feet 6 inches long, fitted with Howden's draft and built for 220 pounds pressure per square inch. The total heating surface is 8,611 square feet and the grate area 194 square feet. The fuel is coal. A donkey boiler, with 1,345 square feet heating surface and built for 125 pounds pressure, is fitted. The transverse coal bunker being oil-tight is fitted also for oil fuel. The engine crew is berthed aft, while the captain and officers are in the forward deckhouse, with the crew in the forecastle. All staterooms have electric light and steam heat. The auxiliaries include one steam windlass, one cargo winch, one mooring winch aft and a steam steerer. Two masts support a wireless telegraphy outfit, and there is also an apparatus for under-water sound signals. 2 plates. 1,270 words.—*Schiffbau,* January 28.

Fuel Oils: Their Origin, Production and Treatment.—By Dr. David T. Day. This article is adapted from lectures delivered in the Post Graduate Department of the United States Naval Academy in February, 1913. It is divided into three chapters dealing with the conditions of occurrence of petroleum, the production of oil and the treatment of oils. Each subject is adequately treated and the article leads up logically to a discussion of the kinds of oil suitable for fuel oils in naval vessels and for use in internal combustion engines, as well as the available supply of such oils. 11,500 words.—*United States Naval Institute Proceedings,* January-February.

S. S. Cap Trafalgar.—The *Cap Trafalgar,* luxuriously fitted as a passenger steamer, has just been completed by the Vulcan Works, Hamburg, for the Hamburg-South American Company. She is 593 feet long and 72 feet beam, with a gross tonnage of 18,000. Propulsion is by two sets of 4-cylinder triple-expansion engines of 6,000 horsepower, each exhausting into a Parsons low-pressure turbine of 7,000 horsepower, driving a center screw. The reciprocating engines run normally at 86 revolutions per minute and the turbine at 210 revolutions per minute. Steam is furnished by Yarrow water-tube boilers, the coal being trimmed by a novel arrangement of vacuum pumps. 6 illustrations. 1,650 words. *The Engineer,* March 20.

Turbine versus Reciprocating Engines for Marine Propulsion.—After discussing the merits and demerits of each type of machinery the author concludes that the turbine is undoubtedly superior to the reciprocating engine as regards economy at high powers, and is therefore almost universally installed on warships, where great speed is required. It is considered, however, that the ideal engine for high-speed warships will be the combination of turbine and reciprocating machinery on account of the gains in economy made at cruising speeds. In high-speed merchant vessels the turbine is found superior, while in moderate-speed vessels either combination machinery, geared turbines or straight reciprocating engines using superheated steam find favor. 2 illustrations. 3,700 words.—*The Marine Engineer and Naval Architect,* April.

H. M. Battleship Queen Elizabeth.—The superiority of the *Queen Elizabeth* in gun power, speed and defensive power over all other battleships afloat is discussed at some length, and then the full particulars of the vessel in comparison with the latest battleships of other nations are given. The *Queen Elizabeth* is 650 feet long, 94 feet beam, and 27½ feet draft. On a displacement of 27,500 tons she carries eight 15-inch guns in four turrets on the centerline of the ship protected by 14-inch armor, and sixteen 6-inch guns behind 8-inch armor along the upper deck and in the superstructure. The main waterline armor is 13½ inches thick, reduced to 10 inches and 8 inches in the strakes above and to 6 inches at the ends. A speed of 25 knots is obtained by Parsons turbines of 58,000 horsepower driving four screws. Steam is supplied by oil-fired Babcock & Wilcox boilers, the bunkers having a capacity of 4,000 tons. 2 illustrations. 2,200 words.—*The Marine Engineer and Naval Architect,* March.

Emergency Repairs to a Destroyer.—By Lieutenant Commander J. F. Hellweg, U. S. N. During the maneuvers off the coast of Cuba last winter the U. S. S. *Burrows* had her bow twisted to port about 80 degrees and badly crushed in collision. The port plating was so badly crushed and distorted that it had to be discarded. The bow was undercut by the blow and the forecastle pulled down and to starboard by the sharp twist of the stem. The vessel returned to Guantanamo Bay under her own steam and was laid alongside a dock for repairs. She was lightened forward to bring her bow out of the water, but it was found necessary to run her bow up on a small marine railway to reach the break in her keel. With the aid of air tools obtained from other ships of the fleet the damaged parts of the bow were removed, straightened and replaced. The forward frames were so badly mangled, however, that they had to be discarded and a makeshift repair resorted to, consisting of oak stringers and breasthooks embedded in cement and secured by through bolts. The entire work was done by members of the crew in 91 working hours, enabling the vessel to resume service under normal conditions. Complete details of the repairs are explained with the aid of photographs and diagrams. 12 illustrations. 2,800 words.—*United States Naval Institute Proceedings,* January-February.

PERSONAL.—Walter C. Allen, general manager of the Yale & Towne Manufacturing Company, New York, has been elected vice-president and general manager of the company.

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New Books for the Marine Engineer's Library

ELEMENTARY MATHEMATICS FOR MARINE ENGINEERS. By J. W. M. Sothern, M. I. E. S., and R. M. Sothern, M. I. E. S. Size, 5 by 7½ inches. Pages, 174. Illustrations, 28. Glasgow, 1914: James Munro & Company, Ltd. Price, 2/6 net.

Intended by the authors as an introduction to the study of mathematics, this book has been specially written for the use of marine engineers who may not have had the opportunity of studying this subject during their earlier education. The subject has been treated from a very elementary standpoint, and the book may be likened to a stepping stone between ordinary arithmetic and mathematics proper. The book is divided into four sections, dealing with elementary algebra, logarithms, temperature entropy diagrams and elements of trigonometry. Those men who have been baffled in attempts to master the subject of marine engineering on account of the mathematical calculations involved therein will do well to secure a copy of this book and make a new start.

NEWCASTLE-UPON-TYNE YEAR BOOK AND COMMERCIAL REVIEW, 1914 (with Sunderland section). Edited by Herbert Shaw, B. A., F. R. G. S. Size, 7¼ by 9¾ inches. Pages, 230. Numerous illustrations. Newcastle-Upon-Tyne, 1914: The Newcastle and Gateshead, Inc., Chamber of Commerce. Price, 2/6.

This is the second issue of a year-book compiled to provide reliable information as to the nature of the extensive industries carried on in the Newcastle and Gateshead district. It includes matter of historical interest regarding the district, and of the doings of the Chamber of Commerce during an existence of nearly one hundred years, and also articles on the business carried on in coal mining and shipping, in iron and steel manufacturing, in shipbuilding and engineering, and in other industries, the chief aim of the book being, of course, to give commercial men in other parts of the world reliable data regarding the resources of this locality and the ability of the manufacturers of all kinds to meet almost any demands made upon them.

TURBINES APPLIED TO MARINE PROPULSION. By Stanley J. Reed, Assoc. M. Inst. C. E. Size, 7¼ by 10½ inches. Pages, 174. Illustrations, 113. New York, 1913: D. Van Nostrand Company. Price, $5 net.

The contents of this volume consists of a special course of lectures given by the author at the Naval Architecture Department of Glasgow University, although in some places the lectures have been amplified. The book is more especially intended for practical use, and for this reason thermodynamics and mathematics are entered into as little as possible. The book is of particular value on account of the thorough and complete descriptions given of the design and construction of the Parsons, Curtis, Zoelly, Rateau, Tosi, Melms-Pfenninger and other types of turbines. The size of the book enables the use of large and clear diagrams and illustrations, many of them being presented in the form of folding plates, so that the details of construction can be shown complete. The titles of some of the chapters given below indicate the wide scope of the book and its usefulness in covering such subjects as have to do with the successful application of turbines to ship propulsion. For instance, there are chapters on the steam thrust of reaction and impulse turbines; balancing propeller thrust; superheated steam, with descriptions of various types of superheaters; cavitation, showing the limitations of the speed of propellers and the various types of reduction gear that can be interposed between the propeller and the prime mover; the relative effect of high vacua with turbines and reciprocating engines; the utilization of auxiliary engine exhaust in the turbines, and the utilization of the auxiliary engine exhaust in feed heaters with the relative advantages of the two systems.

MCANDREW'S FLOATING SCHOOL. By Capt. C. A. McAllister. Size, 6 by 8½ inches. Pages, 250. Illustrations, 37. New York, 1914: Aldrich Publishing Company. Price, $2.

Unlike most textbooks, "McAndrew's Floating School" is written as a story. Four ambitious young men who have begun a seafaring life "below the grating" on an Atlantic steamship, decide to work their way up from positions as oilers, coal passers, water tenders and firemen to the coveted post of licensed engineers. They prevail upon their chief engineer to help them out by giving them instruction in the elements of marine engineering. The instruction is given in the form of lectures, in simple everyday language which men in the engine room of a steamship can readily understand. The lectures cover a wide variety of subjects, beginning with explanations of force, work and power, heat, combustion and the generation of steam. The various materials used in marine engineering are taken up so that the young men can fully comprehend the limitations and usefulness of such materials. Starting at the source of power—that is, with the boilers—a very exhaustive description is given of the construction of steam boilers and of their fittings and accessories. Going then to the engine room, everything which goes into the make-up of the complex machinery of a steamship is described in detail, including all the auxiliaries, pipes and valves and electrical apparatus. A lecture is devoted to the subject of indicators and horsepower, and then the ambitious students are instructed in the care and management of boilers, engines and auxiliaries, giving them a chance to apply under actual working conditions the knowledge which they have gained from the earlier lectures. The final chapter contains a large number of questions and answers.

Every young man who is seeking promotion in the engine room on board ship should read this story and study the lectures given to the students in McAndrew's Floating School." The author of the book, Capt. C. A. McAllister, Engineer-in-Chief of the United States Revenue Cutter Service, is a well-known writer on marine engineering subjects, and in this book he has entered into the spirit of the story as can be done only by a man whose personal experience has brought him into intimate contact with every detail of the subject of which he is writing. As stated by the author in the preface: "The men 'below the grating' of a steamship are the backbone of modern seafarers, as they bear the heat and burden of the work of transportation. Inured from early youth to the hardest physical toil known to man, it is scarcely to be presumed that their early education is such as to fit them for the higher positions in engineering." It is to help men of this type that the book has been written, and the author pays further tribute to his readers by saying: "Certainly these men, the oilers, coal passers and water tenders and firemen of the world's merchant marine are deserving of the efforts of all who can be of assistance to them. The transformation from sail to steam on the great oceans of the world has been so rapid that poets and writers generally are still singing the praises in song and story of the old-time mariner, and have not awakened to the fact that the sailor has been replaced by the man in the fire-room or engine-room. Some of the romance of the sea has disappeared with the sailorman, but heroism is as much and more in evidence with his begrimed successor. Where in the annals of seafarers was ever shown greater heroism than that displayed by the engineers' force of the ill-fated *Titanic*, who almost to the man sank with their ship and at their posts of duty? All the world knows that these hundreds of brawny artisans of the deep could have seized the boats and escaped with their lives; but they were men, and it is for the thousands of men like them that this book was written."

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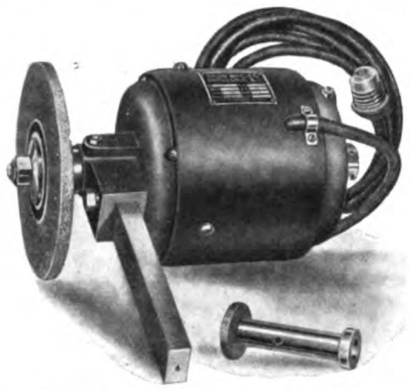
ENGINEERING SPECIALTIES

Electric Tool Post Grinder

For grinding centers, dies, cutters, etc., and also for both internal and surface grinding, the Stow Manufacturing Company, Binghamton, N. Y., supplies an electric tool post grinder fitted with a ⅜-inch by 6-inch emery wheel and an internal



grinding attachment. The tool is attached by a flexible cord to any standard lamp socket and can be furnished to operate on 110 or 220 volt 60 cycles alternating current or 110 or 220 volts direct current.

The Use of the Non-Return Valve in Preventing Accidents

The most important uses of the non-return valve are to equalize the pressure between the different units in a battery of boilers, and to prevent the flow of steam from traveling in a reverse direction to its normal flow.

In case a tube blows out in a boiler, the non-return valve closes automatically, owing to a reduction of pressure, and prevents the header steam from entering the boiler. It acts also as a safety stop to prevent steam being turned into a cold boiler when men are working inside, because it cannot be opened when there is pressure on the header side only.

To be successful, a non-return valve should not open until the pressure in the boiler is equal to that in the header. It should not stick and become inoperative; it should not hammer or chatter while performing its work, and it should be so designed that wire-drawing will not cause wear on the seat and the resulting leak.

In adopting these fundamental principles of safety necessary for the operation of a non-return valve, the Nelson Valve Co., Philadelphia, Pa., have added many additional operating and construction details to the Nelson Cushioned Non-Return Stop Valve, which are interesting from an engineering standpoint, and necessary for reliable operation.

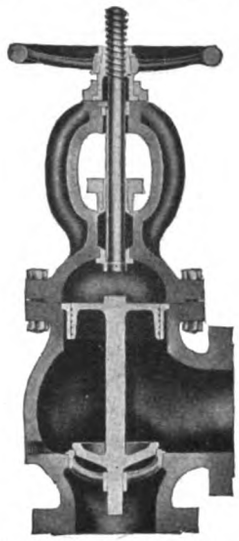
The valve is so constructed that it operates automatically, like an ordinary check valve, and it can be used as a stop valve. It contains an internal dash-pot, having the full area of the valve opening, which acts to cushion the effect of opening and closing the valve. The dash-pot is always at the same temperature as the other working parts, so that contraction or expansion of all parts will be equal, and there can be no binding of the piston; it is, therefore in condition to operate uniformly under all conditions. It is made separate from the body and bonnet castings, thus allowing perfect alinement with the piston.

The piston is made the same depth as the dash-pot, so that in traveling up and down in normal operation there can be no shoulder formed, thereby the cause of sticking, which is fatal to a valve of this type, is removed.

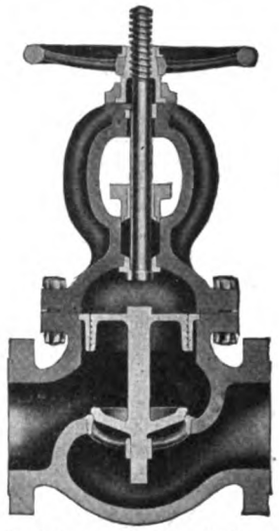
The piston and disk are made in one solid piece of bronze. The disk is provided with a lip below the finished seating surface, which is designed to give an easy flow of steam.

Since a valve of this type must necessarily operate continuously with a short-stroke the damage from wire drawing is considerable, and naturally, with incorrect design, this wear will cause serious leakage. In the Nelson Cushion Non-Return Valve, the lip located below the seating surface absorbs this wear; the seating surface remaining in a smooth condition.

When it is desired to use the valve as a non-return valve, the hand-wheel is opened, as with an ordinary stop valve.



Angle Cushioned Non-Return Valve



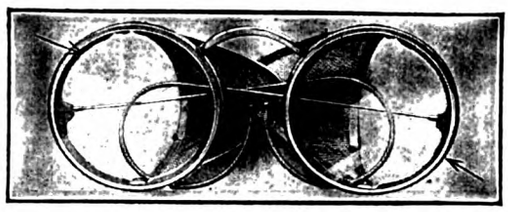
Straightway Cushioned Non-Return Valve

This allows the disk, which is a part of the piston, to operate automatically in the dash-pot with slight changes of pressure. When it is desired to use it as a stop valve, the hand-wheel is screwed down in the usual manner. The hand-wheel is made stationary, so that the valve may be operated with small head room, which feature is often of great importance in many boiler plants.

In the installation of these valves of either the angle or globe pattern, they should be so placed that the pressure is always under the disk, with the stem vertical. They can then be packed when open and under pressure.

Improved Eye Protector for Industrial Workers

Safety goggles, which are claimed to embody new principles in industrial eye protection, have been brought out by T. A. Willson & Company, Inc., Reading, Pa. Instead of the saddle resting upon the bridge of the nose there is an ad-



justable brace bridge designed to distribute the weight evenly upon the sides of the nose and the cheeks.

A safety flange, which is part of the rim, extends over the back edge of the glass to give resistance to blows struck on the lens and holding the glass securely, thus, it is pointed out, preventing injury to the eye from splinters. The wire side screens are either detachable or else are fastened so that they cannot be removed. The shape of the screens is rather unusual, being long and narrow, which, it is emphasized, gives protection without the irritation caused by the edges rubbing and pressing into the cheek and forehead.

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SELECTED MARINE PATENTS

The publication in this column of a patent specification does not necessarily imply editorial commendation.

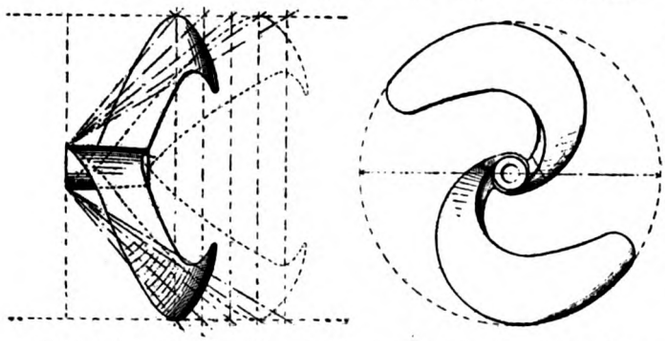
American patents compiled by Delbert H. Decker, Esq., registered patent attorney, Millertown, N. Y.

1,080,625. INTERNAL COMBUSTION PROPULSION APPARATUS. RUDOLPH DIESEL, OF MUNICH, GERMANY, ASSIGNOR TO BUSCH-SULZER BROS.-DIESEL ENGINE COMPANY, OF ST. LOUIS, MISSOURI, A CORPORATION OF MISSOURI.

Claim 1.—An internal combustion engine plant comprising two divisions, an air compressor arranged for operation by one division and propelling mechanism adapted for operation by the other division, in combination with suitable means for driving the latter division by means of the air from the compressor, and means for combining both divisions to operate the said propelling mechanism. Seven claims.

1,087,203. PROPELLER. JULIUS WILLIAM WALTERS, OF GLENS FALLS, N. Y.

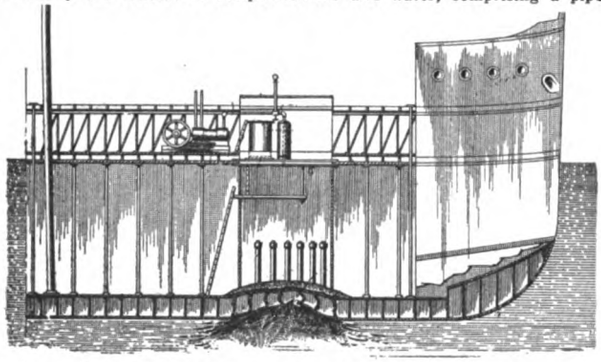
Claim.—A propeller comprising a hub and blades extending outwardly and rearwardly from the hub along a conical spiral curve through approximately 180 degrees to a predetermined maximum diameter, and thence along a helical cylindrical curve of said maximum diameter less than 45 degrees, whereby to prevent overlapping in end projection of the blades, said blades increasing in width from the hub to the point of maximum diameter, thence decreasing in width to their



ends and decreasing in thickness towards their ends, the working faces of the blades commencing tangent to the hub, whereby to admit of free radial flow of the fluid from the hub and offset vortexing, said working faces continuing outwardly and backwardly at an increasing inclination to the transverse plane of the hub, the rear edges of the blades lagging behind the cutting edges up to the point of maximum diameter, and thence being uniformly inclined, whereby the propeller is given worm action effecting the movement of the fluid in a substantially solid column, the outer cylindrical portions of said blades acting upon the solid column of the fluid. One claim.

1,089,967. METHOD AND APPARATUS FOR FLOATING AND REPAIRING STRANDED VESSELS. HERBERT B. SAUNDERS, OF NEW YORK, N. Y.

Claim 3.—A device for confining to a restricted area pulsations caused by the contact of compressed air and water, comprising a pipe



having a flanged base provided with a plurality of apertures, spacing elements arranged in said apertures for holding said base spaced from its support, and an apron disposed between said base and the support on which said base rests. Five claims.

1,081,843. INDICATOR. LAWRENCE S. LARSON, OF TOLEDO, OHIO.

Claim 1.—In a ship's ballast indicator, the combination with a ballast tank, of a float vertically movable therein, an electrical circuit including a volt meter, a bank of resistance elements serially connected to one terminal of the circuit, a contact member rigidly connected to and actuated by the float, a flexible connection between the opposite terminal of the circuit and the contact members and means to insure engagement of the contact member with one resistance element at all times. Two claims.

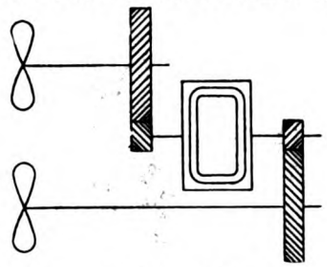
1,081,679. ATTACHMENT FOR LIFE-BOATS. CHARLES H. LANGILL, OF BOSTON, MASSACHUSETTS.

Claim 1.—In a device, the combination of a supporting member including an eye; a telescopic supporting member; a downwardly extending hook pivoted thereto adapted to engage said eye, said hook having its pivotal axis in the same plane with said hook; and a weight adapted to turn said hook about its pivotal axis to release said eye. Eight claims.

British patents compiled by G. E. Redfern & Company, chartered patent agents and engineers, 15 South street, Finbury, E. C., and 10 Gray's Inn Place, W. C., London.

18,977/1913. SYSTEMS OF MARINE PROPULSION AND APPARATUS THEREFOR. W. J. ROSS, 52 BARNS STREET, CLYDEBANK, GLASGOW.

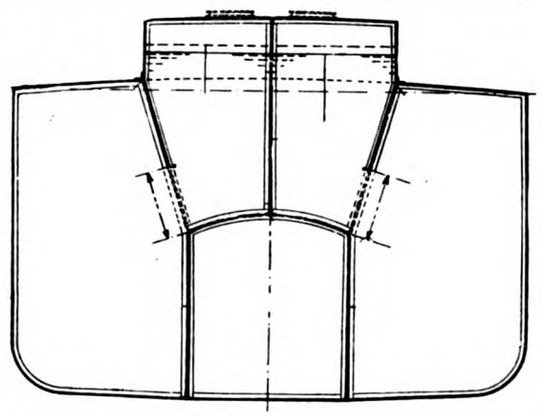
Claim.—This invention relates to the application of radial-flow turbines, of the type having oppositely rotating members within the same casing, to the propulsion of marine vessels, and comprises installations having two or four shafts, two shafts being driven by one turbine of



the radial-flow type having oppositely rotating members, each of which drives one shaft; or in installations having one or more shafts, driving the shafts independently by a radial-flow turbine having oppositely rotating members, one of which is geared direct to the shaft, and the other geared to the same but driving through an intermediate wheel. Either of these arrangements may be used as a cruising turbine, as an independent turbine or in series with the main turbines.

24,741/1912. IMPROVEMENTS IN OR RELATING TO TANK SHIPS OR VESSELS. D. BLACK, of 88 PARK ROAD, NEWCASTLE-ON-TYNE.

This invention relates to the construction of tank ships, for the transportation of liquids or semi-liquids in bulk. A central tank is provided for the storage of water ballast when oil is not being carried, and is equipped with an expansion trunk. Adjoining this chamber are side



tanks also equipped with expansion trunks. The extended sides of the deck trunk are formed with air holes to lighten the structure, while wash plates are provided in the expansion trunk of the central oil hold, and longitudinal oiltight bulkheads are arranged, respectively, between the side oil holds, and in the lower part of the oil holds.

19,780/1912. IMPROVEMENTS IN AND RELATING TO SYSTEMS OF SHIP PROPULSION. THE BRITISH THOMSON-HOUSTON COMPANY, LIMITED, 83 CANNON STREET, LONDON, E. C.

This invention resides in a method, in a system of ship propulsion in which a plurality of propeller shafts are employed having turbines or turbine stages, mounted on each, of operating at slow or cruising speeds, which method consists in supplying steam only to turbines on some of the shafts, the other shaft or shafts remaining idle, and in allowing the steam to expand to practically the same extent or in the same number of stages as when working at high speed. The intermediate stages or pressure turbines on the idle shafts are by-passed and the steam caused to pass directly from the higher stage to a lower stage on one of the operating shafts.

20,095/1912. IMPROVEMENTS IN CHANGE SPEED GEARING, SPECIALLY ADAPTED FOR SCREW PROPELLERS AND OTHER MECHANICAL DEVICES WHERE END THRUST EXISTS. J. MONAGHAN, 17 ORME STREET, BLACKPOOL, AND OTHERS.

This relates to a change speed driving mechanism (for screw propellers and other mechanical devices where end thrust exists) of the type in which the first motion and driven shafts are placed in line with the end of one abutting against the end of the other, and in which these two shafts are geared together near the place where they abut by a counter shaft and differential gear. By this invention the differential gear wheels are attached to their respective shafts, and the counter shaft can be raised to bring its gear wheels out of mesh with those of the driving and driven shafts. Means are provided for coupling the driving and driven shafts together when the counter shaft is raised out of gear, or uncoupling them when the counter shaft is lowered into gear, substantially as described.

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